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EVALUATION OF CG SPILLED OIL RECOVERY SYSTEM (SORS) FOR CURRENTS GREATER THAN 2 KNOTS

FINAL REPORT

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N O T I C E

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16. Abstract (MAXIMUM 200 WORDS)

The purpose of this project was to develop a flexible skimming system, based on the submergence plane concept and utilizing a hydrofoil to counter submergence plane rise, for use alongside a U.S. Coast Guard buoy tender. The system employs the existing Fast Sweep inflatable, conventional oil boom as its aft perimeter component. This report describes system development, hydrofoil design, and 1/5 scale and 1/2 scale physical model testing and full-scale design.

The 1/5 scale physical model was evaluated in Froude-scaled tank and field tests (without oil) at speeds up to 10 knots (full-scale) and waves up to 6 ft (full-scale). Results showed submergence plane rise to be well within acceptable limits with no diving tendency or other evidence of dynamic instability. The 1/2 scale model was tested at Ohmsett where it towed level with no submergence plane rise at the top carriage speed of 6.5 knots (9.2 knots full-scale). Nearly all heavy oil was retained at speeds below 2 knots, while 61 percent of the oil encountered was retained in flat water at a vessel operating speed of 2.8 knots. Over 30 percent of the oil was retained at 4.2 knots. This system shows improvement over existing systems in oil retention at speeds that make it easier to operate vessels. A full-scale version may perform even better although some deployment and handling issues would need to be addressed.

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EXECUTIVE SUMMARY

The existing system for cleaning up spilled oil using Coast Guard buoy tenders, called the Spilled Oil Response System (SORS), only operates effectively up to 1-1/2 knots. This low speed makes it difficult for the vessels to maneuver. The purpose of this project was to develop a flexible skimming system based on the submergence plane concept and utilizing a hydrofoil that can permit the vessels to collect oil at higher speeds. This will increase the area covered as well as permit the vessels to safely maneuver. The system makes use of the existing Fast Sweep inflatable, conventional oil boom owned by the USCG as its aft perimeter boom component. This report describes the hydrofoil design, Hydrofoil/Fast Sweep system development, 1/5 scale physical model testing, half-scale model testing at Ohmsett with oil, and full-scale design.

Submergence plane, flexible barriers, recently developed at the University of New Hampshire, have shown an ability to contain oil at speeds 2 - 3 times that of conventional boom. In operation, oil is forced down the forward submergence plane and enters a sheltered containment region through a gap. Fluid velocities in the containment region are slower, so an aft perimeter barrier can retain the oil. A previous study, however, showed that the submergence plane rose towards the surface at higher speeds negating its effectiveness. In this project, submergence plane rise is countered using a downward acting hydrofoil mounted beneath the submergence plane.

After preliminary trials using a scale model of a related system that had been constructed in a previous project, the forward submergence plane was designed, and a 1/5 scale physical model fabricated. The model was completed using a scaled Fast Sweep aft barrier and a horizontal, fabric baffle across the base of the containment region. Hydrofoil design was based on model measurements of the downward force required to counter submergence plane rise and the wealth of airfoil/hydrofoil theory and cross-section test results available in the literature. The first foil design held the system down well at low speeds, but generated too much force at high speeds causing the bow to dive. The second foil design provided the desired downwards force. The physical model with the improved foil was evaluated in Froude-scaled speed tests without oil up to 10 knots (full-scale) and wave tests up to 6 feet (full-scale). Results showed submergence plane rise to be minimal at the 5 knot upper limit of the specified oil recovery operating range, and no diving tendency or other evidence of dynamic instability was observed. Open water tests alongside a 30-foot towboat, serving as a scaled buoy tender, also showed stable, straight running, maneuvering and towing behavior at Froude-scaled speeds up to 10 knots (full-scale). The conclusion was that the hydrofoil concept had been well demonstrated by physical model testing at 1/5 scale.

A half-scale (14 foot wide) physical model was constructed and tested at Ohmsett, the National Oil Spill Response Test Facility. The model towed level with no submergence plane rise at the top carriage speed of 6.5 knots (9.2 knots full-scale). Nearly all oil was retained at speeds below two knots, while an average of 61 percent of the oil encountered was retained in flat water at a convenient vessel operating speed of 2.8 knots. The system retained about 35 percent of the oil at 4.2 knots.

The half-scale marine aluminum frame was analyzed structurally using strength of materials and finite element approaches. A final design for a full-scale (28 foot wide) prototype was completed and drawings prepared.

The half-scale system could be utilized as built to mount on smaller vessels to increase their oil collection capability. It is expected that a full-scale system will retain more oil, since the gap opening is larger and the containment region is longer and deeper which decreases the probability that the oil will escape. Recommendations are provided for developing a full-scale system.

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I. INTRODUCTION

Purpose and Scope

The use of vessels to mechanically "sweep" up spilled oil is constrained due to the entrainment of oil beneath conventional boom that occurs when the perpendicular component of the current exceeds one knot. This speed limitation creates two significant inefficiencies: very few vessels are capable of maintaining steerage at slow speeds and the surface area that a vessel can sweep in any given period is greatly reduced. The overall goal of this project was to develop a flexible skimming system, based on the submergence plane concept and utilizing a hydrofoil, for use alongside a U.S. Coast Guard (USCG) buoy tender. The system makes use of an existing Fast-Sweep boom as its aft perimeter barrier. In this study submergence plane technology was combined with an existing USCG vessel of opportunity skimming system (VOSS) to increase retention at higher speeds.

System development made use of physical scale models tested at the University of New Hampshire (UNH) and at Ohmsett, the National Oil Spill Response Test Facility. The design includes a hydrofoil to counteract lift forces on the inclined submergence plane. Results of the experimental program are incorporated in the final design, which was also analyzed structurally. Dugan (2000) provides additional design details and supporting calculations for hydrofoil development and Noarse (2000) describes additional details on system design and Ohmsett testing.

Background

Flexible barriers developed at UNH have recently shown an ability to contain oil at speeds 2 - 3 times that of conventional oil boom (Swift et al., 1995, 1996a, 1996b, 1998, 1999, 2000a, 2000b; Coyne 1995; Steen, 1997). Figure 1 illustrates a flexible barrier undergoing testing at Ohmsett. In operation, oil is carried down the forward submergence plane and enters the containment region through a gap. Fluid velocities in the containment region are slower than the incident current, so the oil is held in place. Excess water coming in through the gap escapes beneath the oil through exit holes in the horizontal baffle.

Figure 1. 1997 Bay Defender prototype in action at Ohmsett.

During Summer 1998, UNH developed a large, full-scale system employing a NOFI Vee-Sweep (here after referred to as Vee-Sweep) inflatable boom as its aft barrier (Steen et al., 1999). This system was designed for deployment alongside a USCG vessel to collect oil in sweeping operations. Accumulated oil could be pumped from the containment region as it builds up. First, a 1/5 scale model was built and towed (without oil) in UNH's 120 ft long by 12 ft wide by 8 ft deep tow tank to verify that, while flexible, the structure would take the desired shape. The 1/5-scale model test showed a satisfactory deployed configuration at the design speed of 1 - 5 knots (full-scale). In building the full-scale prototype, the intermediate longitudinals were made much lighter to reduce costs and their impact on logistics.

When tested at Ohmsett, the submergence plane rose to the surface destroying the cross-section shape. At speeds above 1 1/2 knots, the intermediate longitudinals were no longer holding the submergence plane down. Instead, they would rise vertically, then lose stability and fall down horizontally. An additional support structure was built down from the tow bridge to hold the downward incline of the submergence plane. Oil containment of the artificially maintained cross-section was, nevertheless, encouraging. Since the system's draft was 4 ft and Ohmsett water depth was 8 ft, there was also concern that the 50 percent blockage might also have affected the system's fluid dynamics. To address these issues, an open water test was conducted in Sandy Hook Bay using a USCG buoy tender. Though conditions were not ideal (3 - 5 foot seas in 20 knots of wind), the alongside tow confirmed that submergence plane rise was a serious problem. As in the Ohmsett tank, at speeds greater than about 1 1/2 knots, the middle of the submergence plane deflected upward to the surface.

These observations indicated that the design concept had potential for high-speed oil collection if the submergence plane rise problem could be solved. A lightweight device was needed to provide a downward force to the submergence plane. The force should automatically increase with incident velocity and not depend on active (powered) machinery. The simplest solution meeting these criteria makes use of a horizontal, transverse hydrofoil mounted below the submergence plane. Set with a negative angle of attack, the hydrofoil would act to pull submergence plane down. The development of a VOSS system, utilizing the hydrofoil concept in conjunction with a stiffer submergence plane structure, was the main goal of the study described in this report.

Objectives

Specific objectives of the research included:

- Developing a transverse hydrofoil design for holding the submergence plane down in its desired cross-section shape,

- Revising the design to incorporate transverse compression members across the submergence plane to increase rigidity and fabric tautness,

- Adapting the design for use of the Fast-Sweep conventional boom as the aft barrier,

- Testing the new Hydrofoil/Fast-Sweep design in the UNH tow tank and in the field using a 1/5 scale physical model,

- Construct a half-scale physical model for speed, sea-keeping and oil collection performance at Ohmsett,

- Incorporate findings into a full-scale design,

- Perform a basic structural analysis on the full-scale system to verify adequate system strength in the final design.

The steps that were needed to achieve these objectives began before the formal start of the funded project. Preliminary work was done using the 1/5 scale Vee-Sweep model fabricated in the prior Steen et al. (1999) study. The Vee-Sweep model was again tow tank tested to carefully measure submergence plane rise. A hydrofoil was designed, fabricated, installed and tow tank tested based on the results in the tow tank. Lessons learned from these preliminary activities were incorporated into the initial Hydrofoil/Fast-Sweep design completed soon after the start of the USCG funded project in August 1999.

A 1/5 scale physical model of the Hydrofoil/Fast-Sweep system was built based on the initial design. The first hydrofoil design alternative was based on the preliminary Vee-Sweep experience and the extensive body of theoretical and experimental results available in the hydrofoil/airfoil literature. To improve performance, a second hydrofoil design was made and an extensive set of experiments completed without the hydrofoil in order to measure the downwards force necessary to keep the submergence plane from rising. The second hydrofoil was specifically sized to provide this force using lift and drag information provided in the literature. The improved hydrofoil was mounted on the base model and tank tested without and with waves. The final evaluation of the 1/5 scale model configuration took place at sea alongside a small towboat.

Design improvements, materials selection and hardware details were incorporated into the revised Hydrofoil/Fast Sweep design. A model of the design was fabricated at half-scale and brought to Ohmsett for testing. The experimental program at Ohmsett included speed runs (without oil) to assess hydrofoil performance at high velocity, runs with waves to observe sea-keeping, and oil collection runs to measure oil retention performance. The effects of varying containment volume exit area and the vertical position of the hydrofoil were also investigated.

Experimental results provided additional information that was incorporated into the full-scale design and then subject to a structural analysis. Strength of the frame and the hydrofoil were evaluated using strength of materials and finite element approaches. Strengthening of critical members was done to ensure adequate safety factors in the final design.

II. DESIGN CONCEPT

Configuration

The overall configuration of the initial concept is diagrammed in Figures 2 and 3. In order to use the Fast-Sweep boom as the aft perimeter barrier, the size is about 3/4 that of the previous Vee-Sweep version. The width of the mouth is 28 ft (rather than 40 ft), and the draft is 32 inches (rather than 48 inches).

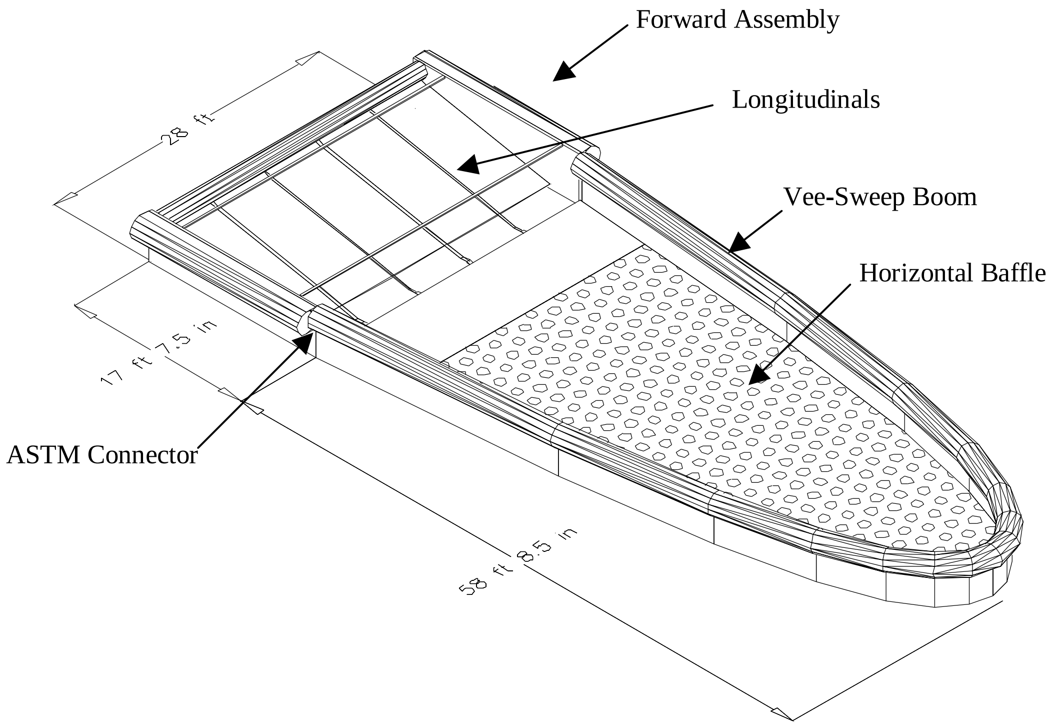

Figure 2. Hydrofoil/Fast Sweep design concept, top view.

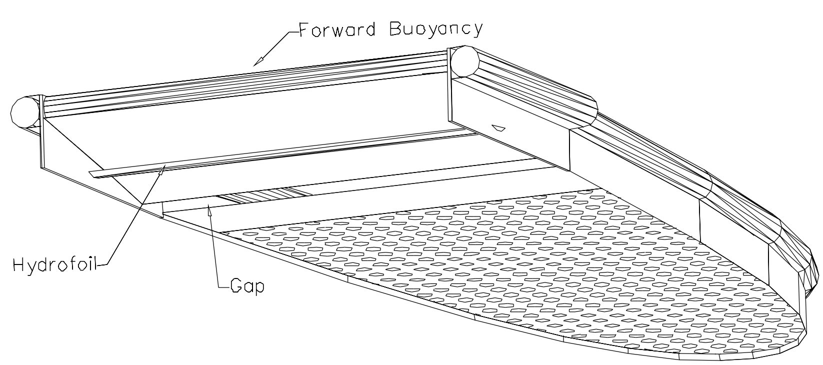

Figure 3. Hydrofoil/Fast Sweep design concept, bottom view.

The hydrofoil is located below the submergence plane in undisturbed incident flow. It is within the profile outline of the end longitudinals for protection while on deck or during launch/retrieval. Its negative angle of attack, cross-section shape and planform area is designed to balance the upwards-planing effect from the submergence plane.

In order to maintain fabric tightness and shape, three transverse compression members (including the hydrofoil) are used to stretch the submergence plane fabric and the gap opening edges. With these new members, shaping of transverse curves to catenary forms (as in the Vee-Sweep version) is not needed. The hydrofoil, leading and trailing edges of the submergence plane and the leading edge of the horizontal baffle are straight and perpendicular to the incident flow. This approach has been previously used in a small scale (12 ft wide) preliminary version designed, built and tested in the Piscataqua River by Weick and Strully (1998) as a UNH senior project.

Another new feature is the ability to separate the system into two sections - the "forward assembly" (submergence plane, longitudinals and forward buoyancy) and the "train" (horizontal baffle/aft barrier portion). The joint consists of two American Society for Testing and Materials (ASTM) connectors with each placed at the trailing, vertical edge of the end longitudinal (see Figure 2). The base of the joint on the end longitudinals includes a hard attachment point for the tension member chain sewn into the leading edge of the horizontal baffle. Thus the structure of the forward assembly is utilized to maintain the critical gap opening geometry.

Intermediate longitudinals are again used to hold the gap geometry and support the submergence plane fabric, but they no longer have an upper structure. Instead, the intermediates are more like battens with an angle change at the gap. The upper structure caused additional weight distribution and handling problems in the Vee-Sweep version, so this portion has been eliminated. There are locking sleeve joints at the gap to facilitate removal of the forward assembly.

Specifications

Based on mission and operational requirements, the following design specifications were established:

- Have a width no less than 28 feet,
- Use the Fast Sweep boom as the aft barrier,
- Use a hydrofoil placed in a protected zone,
- Use transverse compression members across the forward end of the device,
- Use diagonal tension members in the forward end to prevent racking,
- Maintain a submergence plane angle of 11 – 12 degrees,
- Fair the leading edges of the end longitudinals.
- Incorporate adjustable gap dimensions (adjustable when out of water).

The system is intended for operation in at least 3 knots of relative velocity, since this is the minimum speed necessary for vessel steerage. This would permit the buoy tenders to maneuver without using the bow thruster. Additionally, the system should survive towing at faster speeds and in higher sea states. The performance mandates were:

- Survive 10 knots of relative velocity,
- Operate in 3 – 5 knots of relative velocity,
- Operate in 4 – 6 foot seas,
- Tolerate wind loads.

The new system had to be manageable for a buoy tender and crew to on-load, assemble, deploy, operate, retrieve, disassemble and off-load. A site visit was made to the *USCGC Marcus Hanna* a coastal buoy tender stationed in Portland, Maine. The deck plan was obtained, and special deck machinery, such as cranes and winches, were noted. This information was used to plan practical deck storage, assembly, deployment and recovery procedures. It was determined that with all buoyancy deflated, the hydrofoil could be faked down on the aft portion of the working deck with enough space forward for putting together the forward assembly. Using the crane, the forward assembly could then be picked up, attached to the hydrofoil and lowered over the side as the buoyancy is inflated. During recovery, the procedure would be reversed, though it may not be desirable to put an oiled system back on deck if it could be avoided. Returning to a shore cleaning facility with the system alongside is an alternative.

The USCG design mandates were:

- Assembly by 4-6 people within 90 minutes with use of a crane,
- Ability to be launched and recovered from a Coast Guard buoy tender without small boat assistance,
- Minimal system weight,
- Launch and recovery with compression members in place,
- Flotation entirely by air-inflated buoyancy,
- No upper structure to intermediate longitudinals,
- Forward assembly and horizontal baffle / aft barrier to be separable,
- Fabric joints to be ASTM connectors,
- Compression members able to be removed and shortened for storage,
- Disassembly by 6 people in 120 minutes with use of a crane,
- Separate components to be as compact as possible for storage,
- Stored system to fit within a 7-foot by 21-foot area.

III. PRELIMINARY WORK

Tow Tank Testing with Vee-Sweep

Before the formal start of the contract period, preliminary work was done using the existing 1/5 scale model of Vee-Sweep (with submergence plane) fabricated during the prior Steen et al. (1999) project (see Figure 4). Conventional Froude scaling was used for all tow tank tests since this system operates at the surface and forces associated with waves were important. No oil was used during these tests. Thus model speeds were converted to full-scale values using a factor of $5^{1/2}$ and model forces were converted to full-scale by a factor of 5^3 (see appendix A).

Figure 4. NOFI Vee-Sweep Boom Conversion 1/5 scale model fabricated in previous project.

The first set of tow tank experiments was done without modifying the model in order to determine actual, mid-span submergence plane rise. A pointer was attached to the front, mid-span of the submergence plane and read against a reference plane fixed to the carriage. Results were obtained for speeds up to 5 knots (full-scale) where a maximum vertical rise of 27.5 inches (full-scale) was recorded.

Tests were performed to determine how much force would be necessary to hold submergence plane rise down to an acceptable level, defined to be 2 inches (model scale) or 10 inches (full-scale). Known weights were distributed over the intermediate longitudinals and repeating the mid-span rise tests. Though dynamic stability and tank length limitations hampered results for the high-speed tests, sufficient information was gathered to design a hydrofoil that would provide the necessary downwards force.

Hydrofoil Design

It was assumed that, for the acceptable bow rise criterion of 2 inches (model scale), the upward force had the standard quadratic mathematical form,

$$F = \tfrac{1}{2} \rho A C_L V^2 \tag{1}$$

where F is the force (lbs), ρ is density (slugs/ft^3), A is planform area (ft^2), C_L is the lift coefficient and V is speed (ft/s). The coefficient C_L was estimated by fitting the data to the formula. The hydrofoil, having the same velocity dependence, was then selected to balance this force. Using Abbot and Von Doenhoff (1959) as the source for hydrofoil data, a NACA 63_2-615 foil was chosen having a span of 88 inches and a chord of 15.6 inches.

The hydrofoil was fabricated of a thin aluminum skin fastened to plywood ribs and mounted beneath the submergence plane as shown in Figure 5. The hydrofoil ends were attached to vertical plywood plates in line with and extending down from the end longitudinals. Vertical ties were used between each intermediate longitudinal and the top of the hydrofoil.

Figure 5. Hydrofoil mounted under the submergence plane of the Vee-Sweep model.

The hydrofoil restricted submergence plane rise very well at low speeds, but provided too much downwards force at speeds approaching 5 knots (full-scale). Optimizing the angle of attack and foil placement did not alter the basic observation that the foil was oversized. It was also learned that, from the foil design perspective, it is better to have a foil too small rather than too large. Diving of the system bow is potentially catastrophic. These preliminary experiments clearly showed, however, that foil use was technically feasible and that the design methodology was basically sound.

IV. INITIAL HYDROFOIL/FAST-SWEEP DESIGN

Concept Development

An initial Hydrofoil/Fast-Sweep design was developed based on the design concept discussed previously and the preliminary work with the Vee-Sweep physical model. The major components were specified in sufficient detail that a 1/5-scale model could be built for tow tank and open water testing. Choices were made for dimensions and materials of major structural components without working out the many hardware and fitting details. These initial decisions regarding the sizing of parts were based primarily on the experience with the flexible barrier, shown in Figure 1, and Vee-Sweep. Design details and strength calculations were to be completed after model testing.

Forward Assembly

The forward assembly, shown as part of the full system in Figures 6 and 7, makes use of aluminum for the hardstructure. The end longitudinals, illustrated in the Figure 8 cross-section, are framed using 3 inch by 3-inch aluminum box beams. Box sections above the waterline are sealed, while those below are free flooding. The lower bracket structure shown was found necessary in the Vee-Sweep studies to hold the hydrofoil beneath the submergence plane with sufficient clearance.

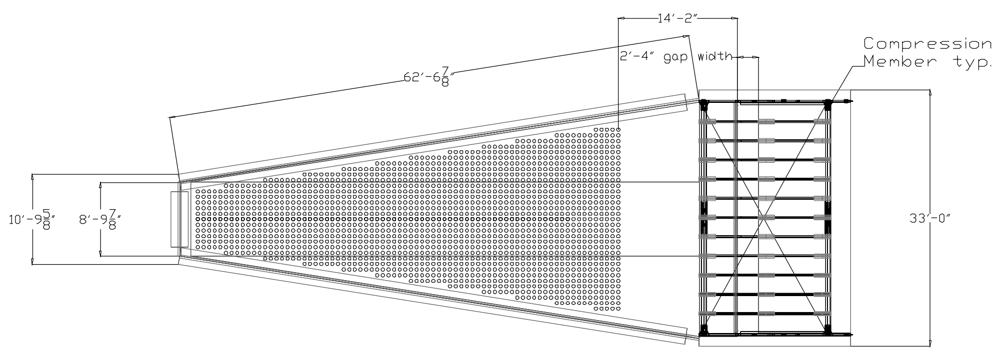

Figure 6. Hydrofoil/Fast Sweep design, top view.

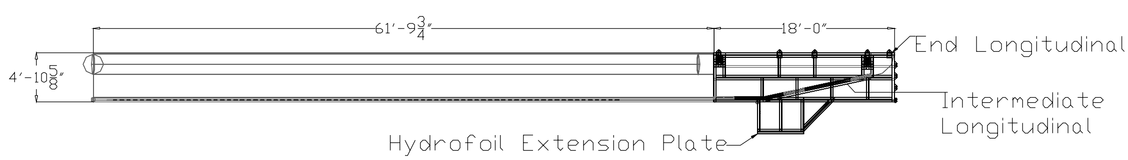

Figure 7. Hydrofoil/Fast Sweep design, side view.

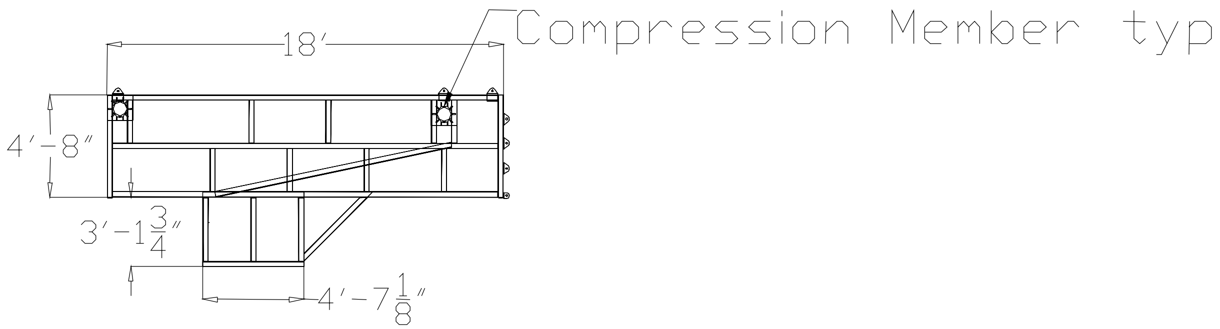

Figure 8. The end longitudinal's frame.

The intermediate longitudinals consist of 2 ½ inch diameter Schedule 40 aluminum pipe, with an angled joint at the gap opening position. The two upper, transverse compression members are 6-inch diameter aluminum pipe. The top diagonal cross bracing is wire cable. The bow and side buoyancy chambers of the forward assembly are air inflatable.

Containment Region

The aft barrier of the containment region consists of the Fast-Sweep boom manufactured by Oil Stop, Inc. As shown in Figure 6, the horizontal baffle is cut to the planform shape of the Fast-Sweep configuration. Exit holes are 5 inches in diameter with an aggregate area equal to 3.6 times the gap area (to reduce exit flow speed and, hence, exit entrainment). A taut, sewn-in chain tension member reinforces the leading edge of the horizontal baffle.

Hydrofoil

The initial hydrofoil design was based on the Vee-Sweep experience assuming the two systems would behave similarly. Thus a NACA 63_2-615 hydrofoil was specified with a smaller planform area taking into account the smaller span (28 ft down from 40 ft) and the observation that the Vee-Sweep hydrofoil model was oversize.

The initial decision for hydrofoil construction was to consider fiberglass. It is the best material for manufacturing small numbers of complicated, curved shapes as well as being reasonably robust in the marine environment and having no corrosion problems. Halves can be female molded then joined, or a shaped core may be covered with a fiberglass skin. The design alternative was marine aluminum, which is stronger. It is in common use by the USCG, and maintenance and repair methods are familiar to USCG personnel. In either case, the hydrofoil would be designed to be neutrally buoyant.

V. HYDROFOIL DESIGN

One-Fifth-Scale Model

A 1/5 scale physical model of the initial Hydrofoil/Fast-Sweep design was built for Froude-scaled testing in the tow tank and open water. Material substitutes were made for ease in fabrication. Specifically, wood was used for a number of aluminum components of the hardstructure. Proper Froude-scaling of mass and weight was accomplished by adding small weights as needed.

The end longitudinals were made using 3/4 inch square pine strips to represent the full-scale aluminum box beam members, while 1/4 inch thick plywood was employed for the foil mounting plates. Lead ballast correction weights were then added such that, when at the designed waterline, the model's end longitudinal net weight was 1/125 that of the full-scale design.

The two upper, transverse compression members were one-inch tubes. Minor discrepancies between exactly scaled model weight and the actual weight of the tube used were taken into account by modifying the ballasting. This approach was used to make small corrections for several other forward assembly components, because as far as the overall dynamics is concerned, the forward assembly frame behaves as a rigid unit. The forward assembly hardstructure was completed using 1/16-inch stranded wire for the cross bracing and is shown in Figure 9.

Figure 9. The forward assembly hardstructure.

Intermediate longitudinals were fabricated using a combination of 1/2 inch copper tube and wooden dowel. The portions were sized so that the combined weight in water equaled the exactly scaled (1/125) weight in water of full-scale intermediates made of aluminum pipe. The intermediates were held in pockets sewn onto the thin waterproof fabric submergence plane. Fabric work used in the model was ordinary, light weight plastic "tarp material". The forward assembly buoyancy chambers were shaped from urethane house insulation foam and covered with fabric. Since they were slightly overweight with respect to air-inflated buoyancy, an appropriate reduction was made in the ballast.

The Fast-Sweep section was manufactured of fabric with encased foam representing the air chambers. The horizontal baffle was fabric having cut out exit area holes. To avoid Reynolds Number effects (excess viscosity impeding the flow), larger, but fewer, exit holes were cut. Total exit area, however, was exactly scaled (1/25) from that of the full-scale design. The assembled Hydrofoil/Fast Sweep model is shown floating in the tow tank in Figure 10.

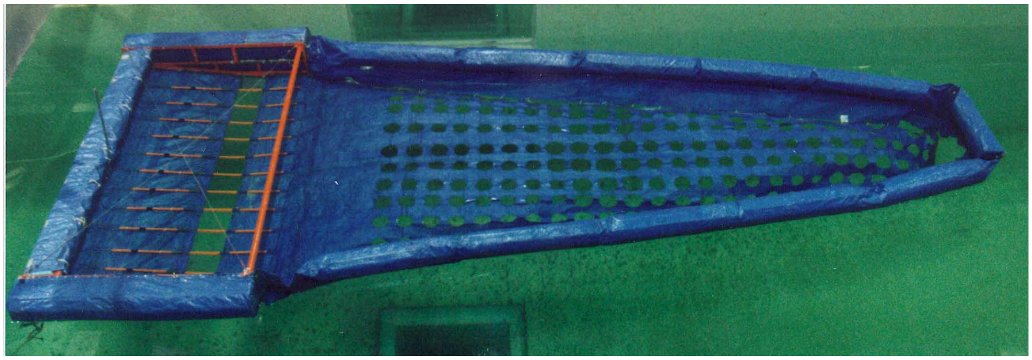

Figure 10. The complete Hydrofoil/Fast Sweep model.

The scale model hydrofoil was constructed of fiberglass over a wood matrix. The internal structure consisted of thin wood planking over plywood ribs. All wood structure was sealed during fabrication. The hydrofoil was then ballasted, covered with fiberglass cloth and epoxy resin, faired and painted. Small holes were drilled top and bottom to make the hydrofoil free flooding, and the ballast distribution was sized so that the final hydrofoil was neutrally buoyant. Figure 11 shows the hydrofoil mounted on the forward assembly beneath the submergence plane.

Figure 11. The hydrofoil mounted under the forward assembly.

Tow Tank Experiments

Tow tank testing quickly showed that the initial hydrofoil design provided more than sufficient downward force for keeping the submergence plane from rising. In fact, it was clearly oversized. The explanation for the improper sizing of the hydrofoil lay in the basic structural difference between the Hydrofoil/Fast-Sweep forward assembly and that of V-Sweep on which the first hydrofoil design was scaled. In the V-Sweep design, each intermediate longitudinal was essentially free to rise individually in response to the planing force. Submergence plane rise was concentrated in the middle away from the end longitudinals, which remained fairly close to their designed waterline. The rigid Hydrofoil/Fast-Sweep forward assembly, on the other hand, rose bodily as a unit. Thus a much greater planing force was required for the same, midspan submergence plane rise.

A second hydrofoil design was needed, so a set of weight experiments were conducted without a hydrofoil to evaluate exactly what the force requirements were for the Hydrofoil/Fast-Sweep model as built. A trough (of known weight) was constructed across the forward assembly, as shown in Figure 12, to hold the imposed weights. With known weights added, the system was towed at 1 - 5 knots (full-scale) and bow rise measured. The data set was then processed to yield downward force as a function of speed necessary to maintain a specified rise. Figure 13 shows the weight force vs. speed results for a bow rise limited to 2 inches (model scale).

Figure 12. The forward assembly with wooden trough.

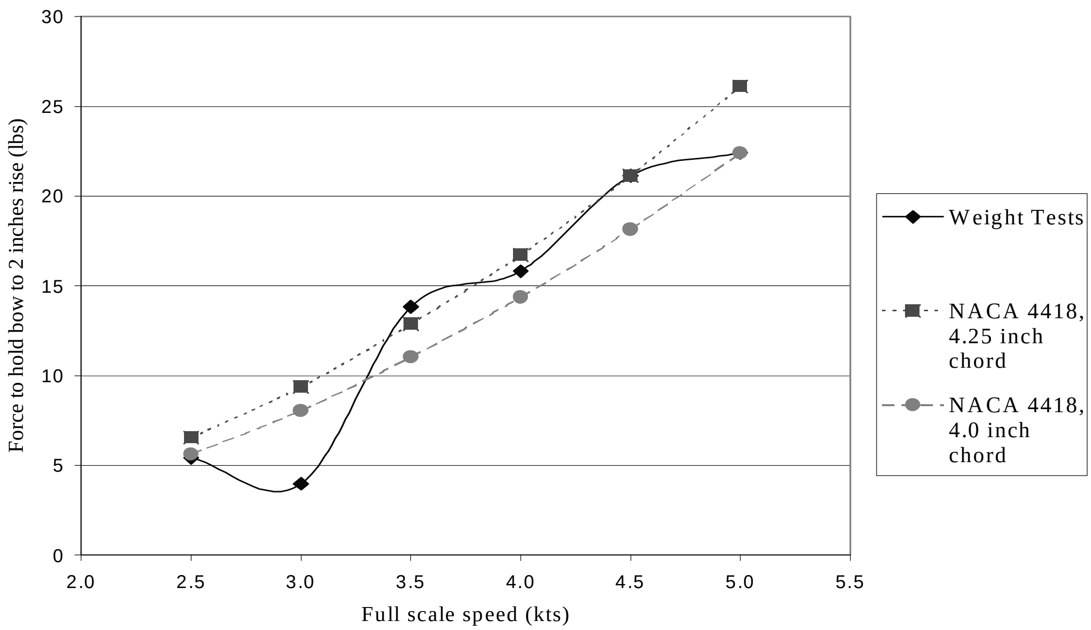

Figure 13. Weight force vs. speed without hydrofoil and hydrofoil force vs. speed for two hydrofoil designs having different chord lengths.

Modified Hydrofoil Design and Fabrication

The design goal was to specify a second-generation hydrofoil that would limit bow rise to an acceptable level but not be so large as to cause catastrophic diving. A conservative approach with respect to hydrofoil size was taken based on the experience with the two previous oversized hydrofoils. Hence, a 2 inch rise (model scale) over the 0-5 knot (full-scale) operating range was considered acceptable, and the weight force vs. speed curve corresponding to this limitation shown on Figure 13 was relevant. Bracketing the imposed weight experimental curve on Figure 13 are hydrofoil force predictions for two full-span, NACA 4418 hydrofoils at 7 degrees angle of attack having different chord lengths. A hydrofoil represented by the lower curve having a chord length of 4 inches (model scale) was chosen in order to keep the bow from diving. This shape, shown in Figure 14, has a relatively high lift coefficient, is somewhat simpler in outline than the previous hydrofoils and is, therefore, easier to manufacture. This hydrofoil will also stall at approximately 10 degrees angle of attack thus putting an upper limit on downwards force should the nose go bow-down.

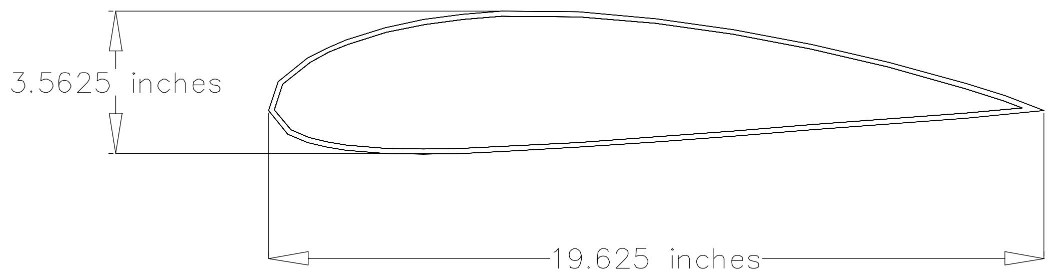

Figure 14. NACA 4418, hydrofoil with full-scale dimensions.

The hydrofoil was shaped from a single clear pine board. Enough lead ballast was added to make the hydrofoil neutrally buoyant. The lead was placed into a longitudinal trough routed into one face. The ballast was then sealed with a strip of fiberglass tape, and the hydrofoil was sanded, painted and mounted on the model.

Bow Rise and Wave Experiments

Physical model experiments with the new hydrofoil demonstrated almost ideal behavior as shown on the Figure 15-bow rise results. In the operating range of 0 - 5 knots (full-scale), bow rise was limited to 1.5 inches (model scale) with no tendency to dive. Tests were then continued at higher speeds resulting in a vertical displacement of 3.25 inches (model scale) at 10 knots (full-scale), again with no loss of dynamic stability.

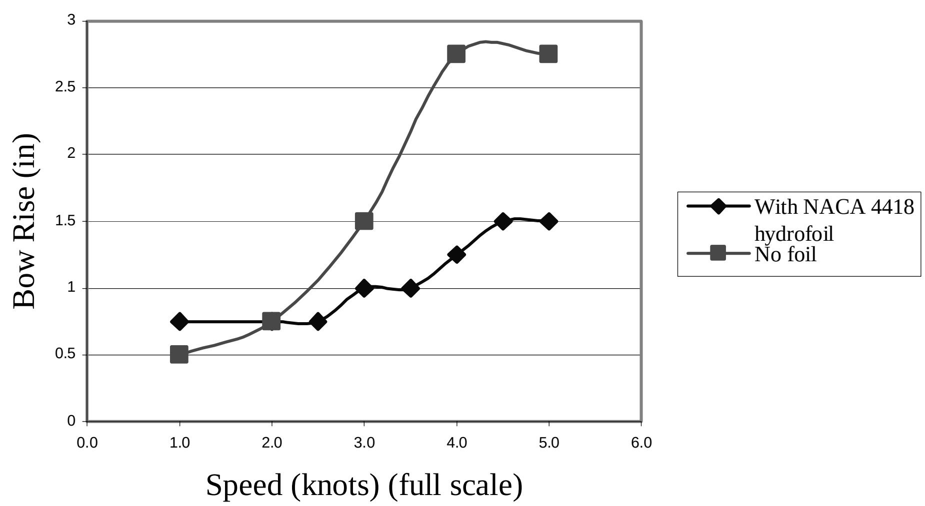

Figure 15. Bow rise with and without hydrofoil.

Dynamic stability was then investigated further in a series of wave tests. The objective was to verify that no surface disturbance would induce a diving mode or any other undesirable dynamic behavior. Three regular (monochromatic) wave conditions were selected for their severity in driving an extreme seakeeping response. Wavelengths used were equal to model full length (14 feet, model scale), 1/2 model full length (7 feet, model scale) and forward assembly length (4 feet, model scale). The wave steepness (wave height/wavelength) was maintained at a severe 1/15 ratio yielding a wave height of approximately 6 feet (full-scale) for the longest wavelength. Tow speeds ranged from 0 - 10 knots (full-scale). While there was spray and slapping of the submergence plane and forward buoyancy chamber at higher speeds, no tendency to dive or otherwise lose dynamic stability was observed.

Tow Force

The tow force applied horizontally to each end longitudinal needed to overcome system drag was measured using spring scales. The towline from the system led through a pulley at the waterline the vertically up to the conveniently mounted scale. The model was again towed at speeds of 1 through 10 knots (full-scale) without waves and then with the three waves used in the seakeeping tests.

Results for the no-waves case are given in Figure 16 where both tow force and speed are Froude-scaled to full-scale. The longest wavelength (equal to system length of 14 feet) caused the highest tow forces as shown in Figure 16. Tow forces for the two shorter wavelength tests were slightly less than the higher curve and are not plotted.

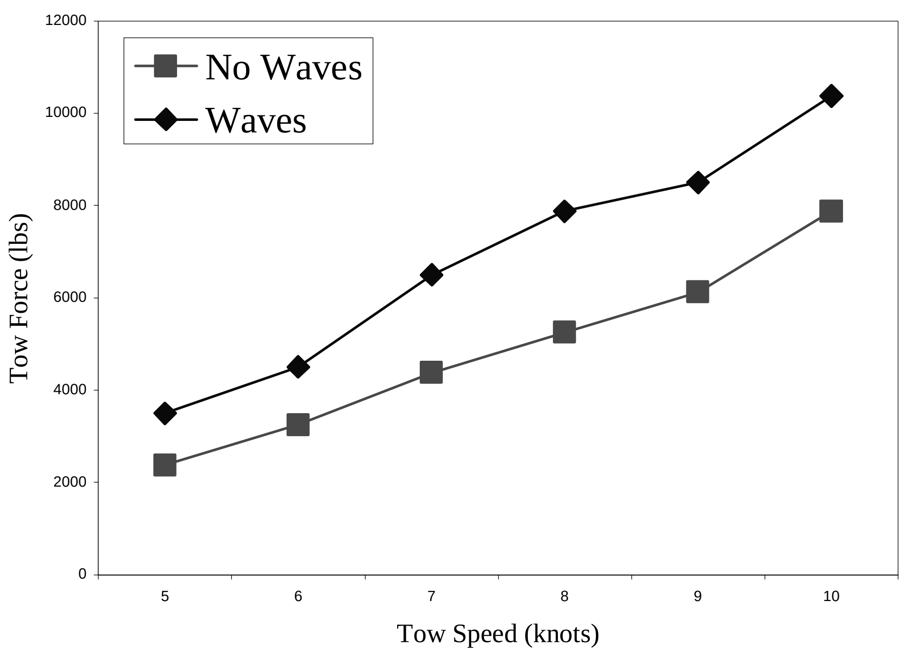

Figure 16. Tow force per end longitudinal results for 1/5 scale model scaled up to full-scale values.

Open Water Testing

The open water experiments were conducted on November 12, 1999 using a 30-foot towboat in the Piscataqua River off the Sprague-Newington oil terminal, New Hampshire. A combination outrigger mount and "artificial side" was clamped to the port gunwale as shown in Figure 17. The "artificial side" represented a scaled buoy tender's topsides and provided the inside towpoint at the proper height. The outrigger was essentially a guyed compression member with the inside mount free to rotate and the outside partially supported by added buoyancy. A forward guy to the bow and an upward line to the cabin top controlled its position.

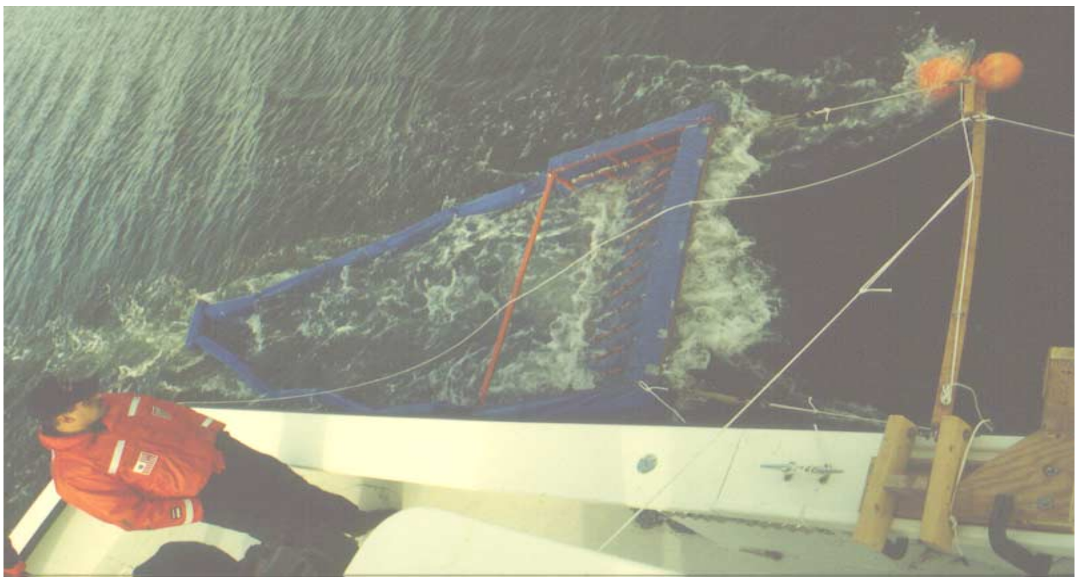

Figure 17. Open water tow test at approximately 10 knots (full-scale).

The Hydrofoil/Fast-Sweep physical model was launched and rigged alongside in a scaled replication of what would be done full-scale on a buoy tender. Speed and maneuvering runs were then made downriver of Sprague-Newington along the New Hampshire shore. Wind was light; wave activity was minimal, and there was a flood current in the area of approximately 1 1/2 knots. Speeds were measured using the vessel's Differential Global Positioning System, corrected for current, and then Froude-scaled to full-scale values.

Overall, the Hydrofoil/Fast-Sweep model performance was excellent over the full range of tow speeds, 0 - 10 knots (full-scale). The system at 10 knots (full-scale) is shown in Figure 17. Once towlines were properly adjusted, bow rise was negligible, and there was no tendency for the bow to dive. Standard maneuvers were accomplished without incident. Turns to the left and right were performed repeatedly at various speeds with no problems.

The system was very sensitive to the height of the tow points. For the final full-scale configuration, care must betaken in maintaining horizontal towlines during deployments to within approximately plus or minus10 degrees. At intermediate speeds there was some tendency for the sides of the containment region to move inwards (that is, narrow the beam of the containment region). Eddying aft of the blunt ends of the side buoyancy may have contributed - a design issue that can be corrected in the next full-scale version. Lateral waves off the side of the towboat may also have affected containment region shape. A transverse crease, or “hinge”, was also noted in the submergence plane fabric between the forward ends of the intermediates and the forward flotation.

VI. HALF-SCALE MODEL

Design Overview

The initial full scale Hydrofoil/Fast Sweep design was updated to produce the next iteration in the system development. Modifications included changes based on the 1/5 scale model experiments. The side flotation chambers were streamlined to reduce back eddying, and the intermediate longitudinals were lengthened to minimize hinging. In addition, the revised design incorporated design details with respect to joints, fittings, structural assembly and air flotation. The half-scale model was a replication of the revised design and was built using materials specified for full-scale construction. The completed half-scale model was then used for speed, sea-keeping and oil retention tests at Ohmsett (as described in the next section).

Detailed design drawings based on the full-scale system were prepared for half-scale model fabrication. The model would be fabricated, assembled, towed and disassembled just as the full-scale system would be. Therefore, its design was much more intricate than the 1/5 scale model. Again, Froude scaling was used to maintain similitude. However, this time, the scaling of component dimensions could also be geometric, since the materials used would be the same as those intended for full-scale design. Also, like the full-scale system, flotation would consist entirely of air instead of foam as used in the 1/5 scale model.

Forward Assembly

The end longitudinals, intermediate longitudinals and compression members form the hard structure of the forward assembly. Discussed individually in the following paragraphs, they are shown assembled in position in Figure 18.

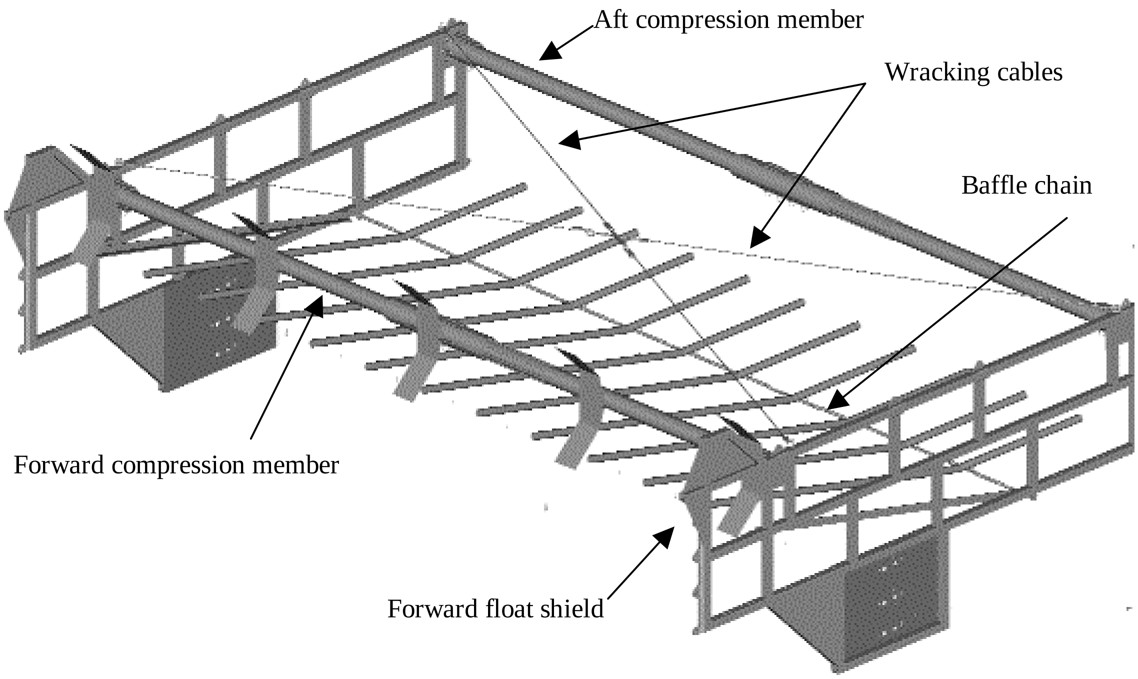

Figure 18. Half-scale aluminum structure of forward assembly.

End Longitudinals

The forward assembly was made from 6061-T6 standard aluminum structural extrusions. Initially, the model end longitudinals were to be made from 11 gage 1 ½ inch by 1 ½ inch square structural tubing. This gage at 0.0907 inches thick would closely conform to a ½ scale thickness, as the intended full-scale structural members were to be 3 inch by 3 inch by 0.188-inch square tubing. However, welding such thin-walled tubing by conventional means risked warping the tubing. Also, such a light tubing structure was not thought to be durable enough in high-speed tests. As a result, 1 ½ inch by 1 ½ inch by ⅛-inch tubing was used for the half-scale model end longitudinal frame shown in Figure 19. This could be welded at half-scale with confidence. However, to maintain similitude, this implied that the full-scale end longitudinals should be made from ¼ inch thick 3-inch square tubing. Upon further consideration, the thickness increase was found desirable for full-scale strength, so the ¼ inch wall thickness was incorporated in the revised full-scale design.

An important modification was use of an air inflated forward float, having a circular cross-section. Since the purpose of the forward float was reserve buoyancy, its centroid was desired to be as high as possible, at least (proportionally) as high as the non-circular forward float of the 1/5 scale model. As such, its centroid would have to be near the top of the end longitudinal, and its radius would need to span from the waterline at the mid-point of the height of the end longitudinal to above the top of the end longitudinal. Since it was inflatable, protection from puncture was required on each side. This was accomplished using the forward float shield shown in Figure 20. This shield was designed to be made from the same aluminum frame stock as the main framing of the end longitudinals, and use ⅛-inch aluminum plating.

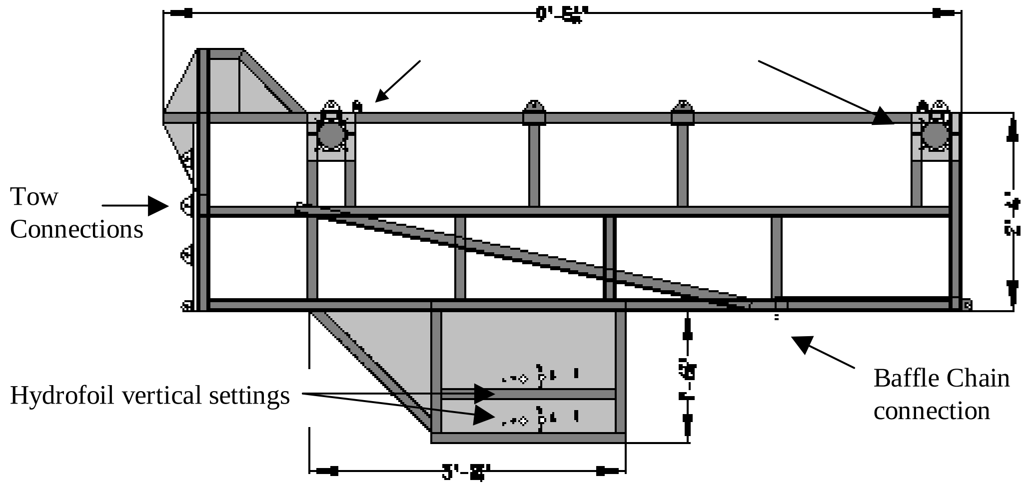

Figure 19. Half-scale end longitudinal design.

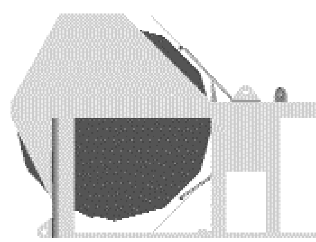

Figure 20. Forward float shield viewed from the side.

Another characteristic of the half-scale model was the ability to change vertical position of the hydrofoil. Influence of the vertical positioning of the hydrofoil in relation to the gap was unknown at this time, but was considered to be possibly critical to oil collection. Two vertical positions were designed into the extension plate (see Figure 21). Each of these positions was aligned horizontally in scaled conformance to the horizontal placement of the 1/5 scale model. It should be noted that since the vertical position of the hydrofoil in the final design depended on the Ohmsett test results, the extension plate design at this stage was provisional.

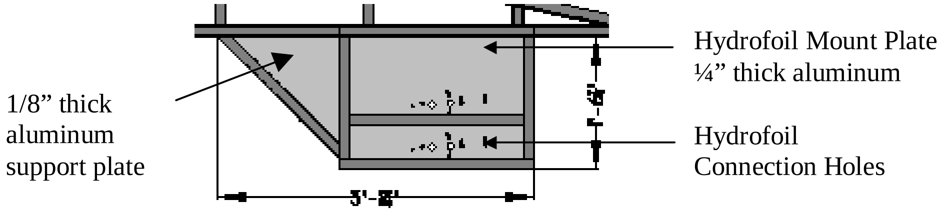

Figure 21. Half-scale hydrofoil extension plate.

Although the sheet aluminum portion of the extension plate to which the hydrofoil attached (hydrofoil mounting plate) was to be ¼ inch thick at full scale, it was made at ¼ inch also on the half-scale model. This is because ⅛ inch thick sheet metal at half-scale was not deemed to have the required durability for a hydrofoil connection with the many holes required (see Figure 19).

Since the waterline would be at the midpoint of the middle horizontal structural member of the end longitudinal, this member and all members above would be airtight. All members below this would be free flooding. This was accomplished by drilling ½ inch diameter holes through each member on 3-inch centers. This spacing was deemed large enough to not compromise the strength of the members and overall frame.

The end longitudinals also had to be further designed to integrate with compression members. Two compression member bases were added to each end longitudinal as receptors for the compression members. These bases were made from reinforced structural piping with angular gussets for support mounted onto ½ inch thick aluminum plating (see Figure 22). Since the half-scale compression tubes would be 3 inch outside diameter aluminum tubing, the base pipes were made from 3½-inch schedule 40 aluminum pipes. With an inside diameter of 3.068 inches, they provided a snug fit for the tubular compression members. Tow connections were made from ½” stock aluminum sheet. Their locations on the forward end of the end longitudinal (see Figure 18) were kept proportional to those of the 1/5 scale model.

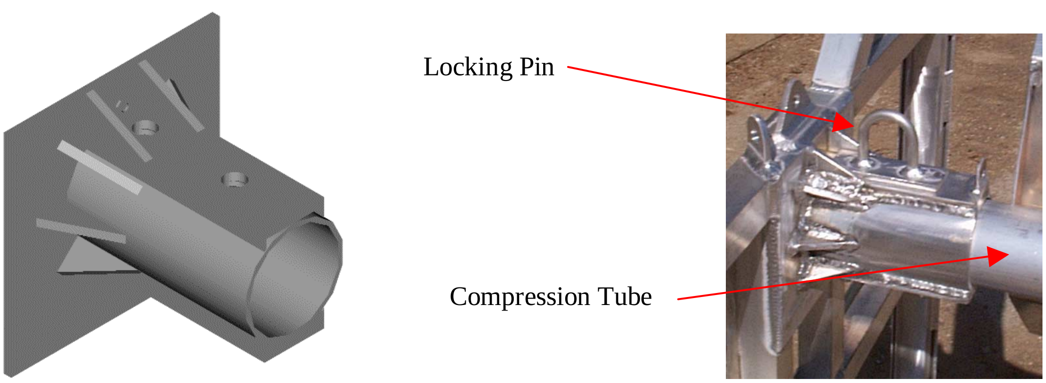

Figure 22. Compression member base, design (left) and fabrication (right).

Intermediate Longitudinals

The full-scale intermediate longitudinals were planned as lengths of 2½-inch schedule 40 pipe with a 12-degree bend at the gap position. For the half-scale model, 1 ¼ inch schedule 40 pipe was used even though each weighed 6.25 pounds rather than the exact Froude-scaled weight of 4.18 pounds. The weight difference was neglected in comparison with the potentially large amount of side and forward buoyancy forces. Also, intermediate longitudinals were designed to be longer than those of the 1/5 scale model to eliminate the hinging effect observed in the 1/5 scale open water tests.

The half-scale intermediates were bent in a jig to a 12-degree angle, conforming to the device's submergence plane. The angle was minor enough so that there was minimal structural weakening of the member. Following the bending, a member was successfully tested for resistance to further bending by placing a 200-pound weight at the bend. An important function of the intermediates was to maintain gap geometry, which was in proportion to that used previously in the Vee-Sweep design and the 1/5 scale model. Provisions were made to adjust the "bite" (the vertical offset between the end of the submergence plane and the level of the horizontal baffle) in case this became desirable during Ohmsett testing.

Compression Members

Each of the two compression members was designed to break down into two 14-foot long full-scale tube sections (7 feet model scale), which would connect at a central junction (see Figures 23-25). Each member would have three components: two tube sections and a center assembly. Assembly of the full compression members meant having to spread the end longitudinals far enough apart to the limit of the submergence plane width. This meant working against the tension of the submergence plane and the tension of the baffle chain. The combined length of each of the two tube sections would, therefore, equal the maximum spread of the end longitudinals. Once each of the two tubes was inserted into the compression base mounts and aligned, the central assembly as a sleeve-like coupling would slide over the tube junction and be pinned into place.

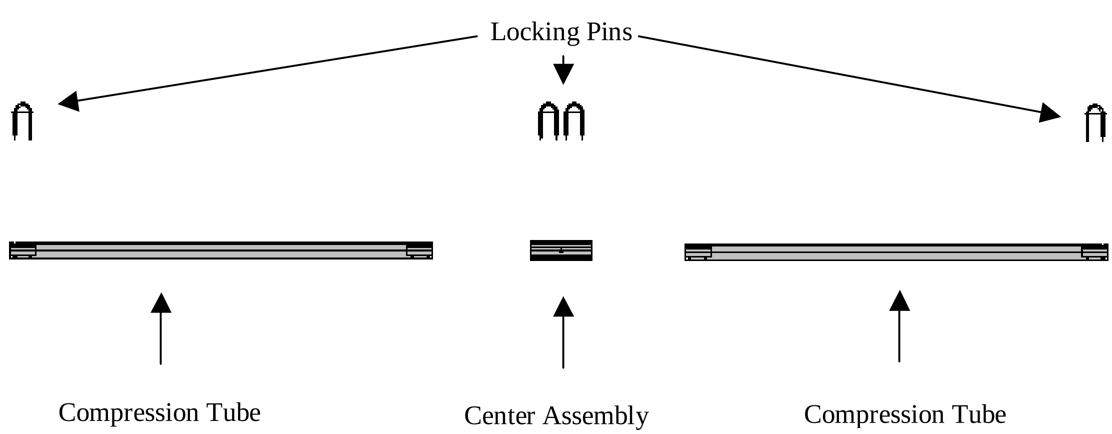

Figure 23. Half-scale aft compression member design.

The half-scale tube sections would be made from 3 inch diameter, ⅛ inch thick aluminum tubing. The center assemblies would be made from 3 ½ inch schedule 40 aluminum pipes like the compression member bases attached to the end longitudinals. Each of the three components would be attached to one another and to the end longitudinals and held in place with aluminum locking pins as shown in Figures 24 and 25. The locking pins were designed to be double-shanked. This way, joints would have fewer tendencies to rotate.

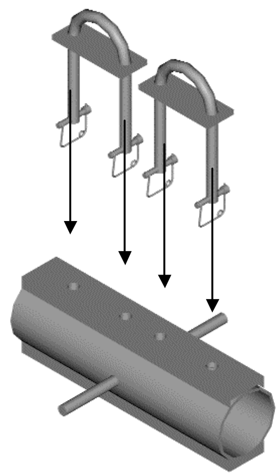

Figure 24. Half-scale aft compression member center assembly.

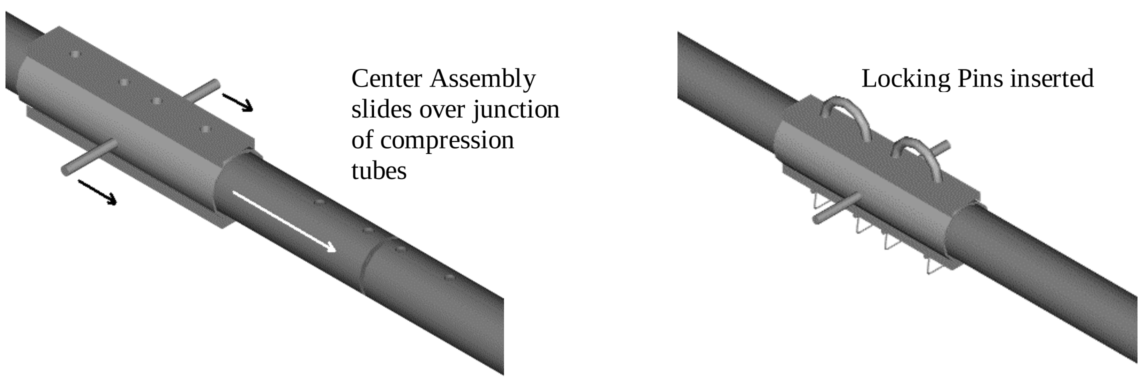

Figure 25. Half-scale compression member center assembly's operation.

Since the forward compression member was located immediately aft of the inflated forward float, it included several attached sheet aluminum brackets. In addition to serving as a strengthener of the forward assembly, the forward compression member also served to support the forward float. The brackets would act as a cradle to support the float and resist bending brought on by drag forces. The half-scale model brackets were made from ⅛-inch thick aluminum sheet and were welded to the forward side of each of the tubes and the forward center assembly as shown in Figure 26. The full-scale brackets would also be made from ⅛-inch thick aluminum since ⅛ inch is adequate at full scale and weight is minimized.

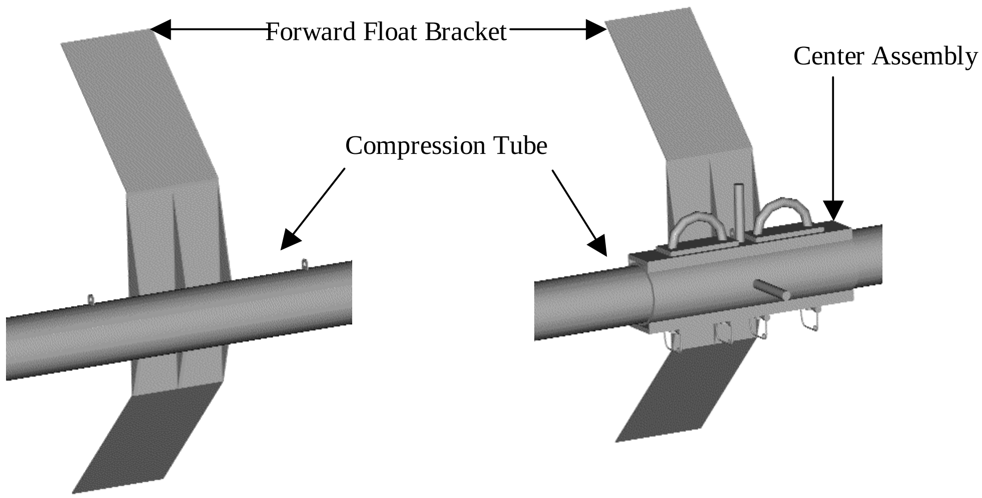

Figure 26. Half-scale forward compression member float bracket (left) with center assembly (right).

Baffle

The half-scale horizontal baffle was made from 28 ounce urethane fabric. To maintain the required exit to entrance area ratio of 3.61, 5- ich diameter holes were cut into the baffle on 7-inch centers. Stainless steel grommets were installed along its 3 boom sides for boom attachment. To reinforce the edges, 2-inch wide canvas webbing was machine sewn to the fabric. The leading edge of the baffle (and aft boundary of the gap) contained an integral chain pocket for installation of the baffle chain. The baffle chain was fed through the baffle pocket. The baffle chain, rated for 3,500 pounds, was a critical structural member of the system. As it helped to maintain gap geometry, it would be under intense tension due to drag forces on the baffle. This tension would be transferred to the end longitudinals. It was connected to connection points on each end longitudinal with a clevis pin. Chain tension would be maximum at the highest system speed of 10 knots full scale.

The tension was calculated by first estimating the form and skin drag forces imparted to the baffle at this maximum speed. Skin drag over the area of the baffle was determined using an empirically-derived, Reynolds Number dependent formula (as given by Owczarek, 1968),

$$C_{SF} = \frac{0.455}{(\log \mathrm{Re})^{2.58}} \quad , \tag{2}$$

where

C_{SF} = Coefficient of Skin Drag
Re = Reynolds Number at the mid-range speed.

The total skin drag was determined using the standard quadratic drag relationship,

$$F_{SF} = ½\rho C_{SF}AV^2 \quad , \tag{3}$$

where

F_{SF} = force (lbf)
ρ = fluid density (slugs/ft^3)
C_{SF} = skin drag coefficient
A = the planform area (ft^2) of the baffle
V = speed (ft/s).

Form drag against the chain pocket was determined using the same standard quadratic form for drag forces, but with a coefficient of form drag, $C_{D.}$ This coefficient was determined to be 1.2 (from Berteaux, 1991). The area used was the projected area of the chain pocket. The combined total drag was calculated as

$$F_T = F_{SF} + F_{FD} \quad , \tag{4}$$

where F_{FD} is the force component due to form drag against the chain pocket.

The combined form/skin drag, F_T per unit length of chain was calculated to be 36.53 lbs/ft. This figure was then used in conjunction with a standard equation for catenary geometry (Berteaux, 1991) to derive the tension in the chain. Since desired chain deformation is small, the presumption is that all drag force is normal to the chain and that chain tension is constant along its length. It was desired to find the chain tension providing 6 inches of deformation due to the drag forces occurred. From Berteaux, (1991), the curve taken by the chain follows the standard catenary formula,

$$y = \frac{T}{P}\left[\cosh\left(\frac{P * x}{T}\right) - 1\right] \quad , \tag{5}$$

where

y = lateral deflection of the chain (ft)
x = one half the chain length (ft)
T = horizontal chain tension (lbs)
P = drag force per unit length of chain (lbs/ft).

The value of T/P is iteratively solved for with y and x established at 0.5 feet and 7 feet respectively. This value is then multiplied by the drag force per unit length of chain previously calculated:

$$T = \frac{T}{P} * \frac{F_T}{L} \quad , \tag{6}$$

where

L = length of chain (ft)
T = total chain tension (lbs).

Calculations revealed that at the top end survival speed of 10 knots (full scale), the half-scale model chain would be in nearly 1,800 lbs of tension due to drag forces on the baffle. This value is well under the chain working load limit of 3,500 pounds.

Submergence Plane

The submergence plane was made from the same 28-oz. fabric as the baffle. It was also reinforced on all sides and beneath each intermediate longitudinal pocket with canvas webbing. It was fastened to each end longitudinal along their angle members (see Figure 27) using a stainless steel bolt. The submergence plane fabric wrapped around the forward float and was secured to the forward compression member with stretch chords (see Figure 28). The intermediate longitudinals were placed into the submergence plane by inserting each one into three fabric sleeves sewn to the plane.

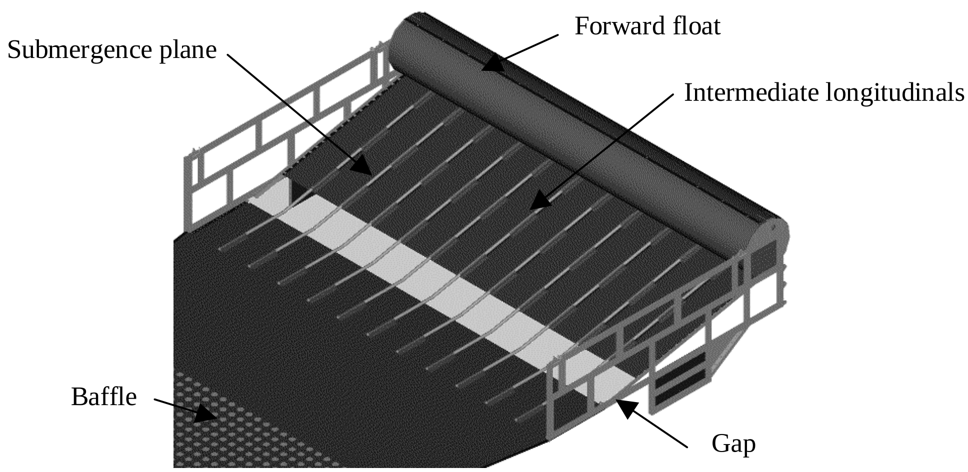

Figure 27. Half-scale submergence plane.

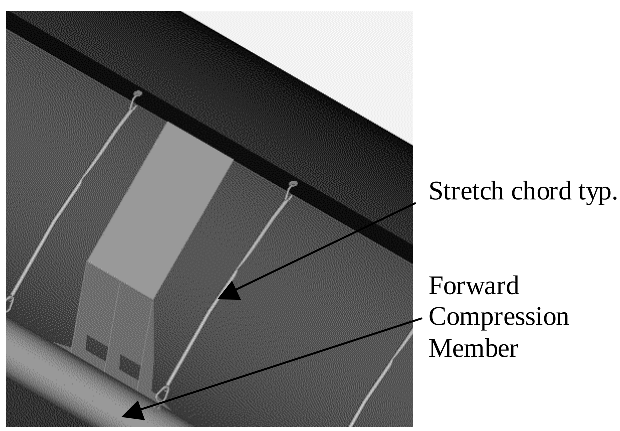

Figure 28. Half-scale submergence plane and forward float.

Flotation

Flotation was supplied entirely by air-inflated buoyancy. The forward float was cylindrical in shape with a half-scale model diameter of 21 inches and a length of 13.75 feet. The 21-inch diameter permitted its cross-sectional centroid to be at the top level of the top length member of the end longitudinals when in position. Its large diameter meant that it would have even more buoyancy in proportion to the 1/5 scale model. Its bottom had a draft of about 2 inches. It was made from 41/40 Seaman fabric manufactured from the copolymer elvaloy. It contained two fill valves, one at each end for easy access.

The side floats were designed to be mainly reserve buoyancy, with about 2 inches of draft. However, due to the back eddy and wake effects observed during the open water trials of the 1/5 scale mode, it was decided to make the forward and aft ends of the side floats more stream-lined and less blunt (see Figure 29).

Figure 29. Side float attached to the outside of the starboard end longitudinal.

The ends were made conical, with the side of the cone bordering the end longitudinal flat against it. It was important that since the ends would be conical, the overall float retain proportional buoyancy, or at least as much as the 1/5 scale model had. This was achieved by the side floats having an additional inflation chamber between the main cylindrical float and the backing fabric (see Figure 30). The half-scale model diameter over their straight length at 30 inches was in proportion to that of the 1/5 scale model. The side floats were made from the same material as the forward float and were fused to a fabric backing of like material, which attached the side floats to the end longitudinals. This backing was attached to 0.08-inch thick aluminum tabs welded to each end longitudinal at intervals with through bolts and machine screws.

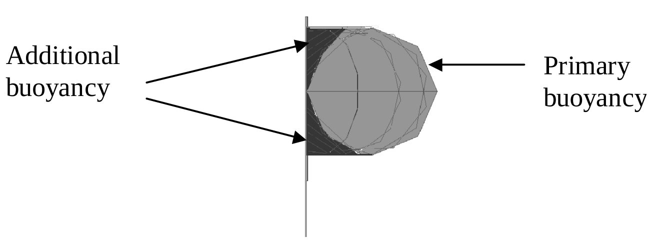

Figure 30. Cut-away section of side float.

Hydrofoil

The hydrofoil was designed to be made completely of 6061-T6 aluminum. A fiberglass option was explored, but found to be cost and time prohibitive. Also, the aluminum option was thought to be much more durable and easily repairable by USCG buoy tender personnel. Figure 32 shows the basic constitution of the hydrofoil. The complete description of hydrofoil development is provided by Dugan (2000).

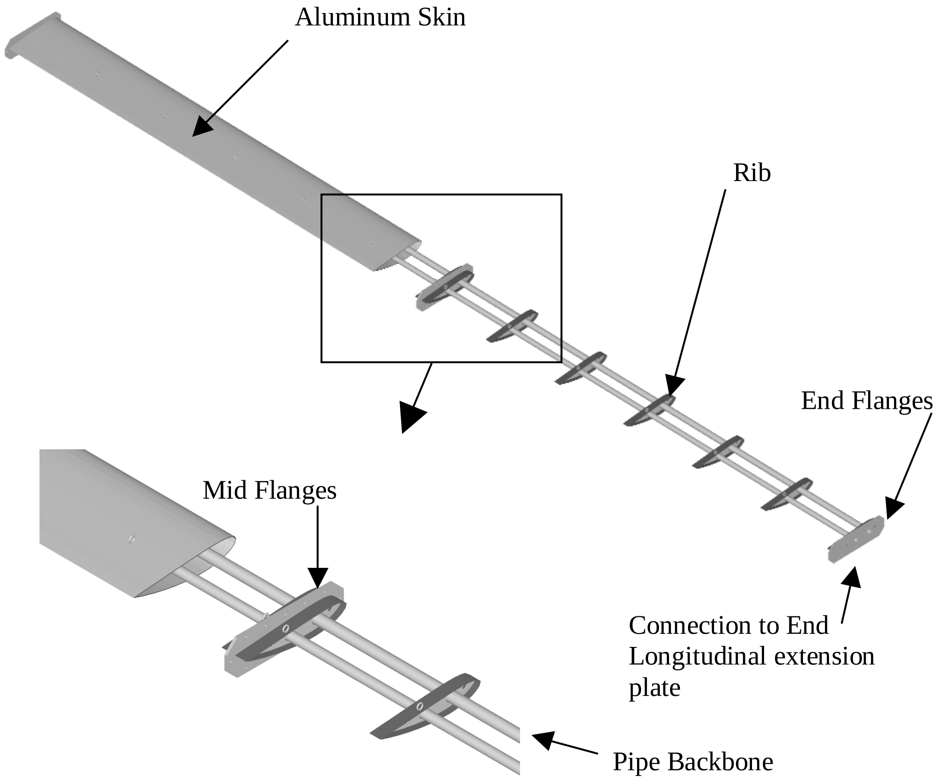

Figure 31. The half-scale aluminum hydrofoil design concept with skin partially removed to show detail.

As seen in Figure 32, the hydrofoil was designed as a structure having aluminum ribs mounted on two longitudinal pipes and covered with an aluminum skin. The hydrofoil is separable into two halves with each end terminated in an exterior flange. The flanges, in turn, were bolted to each other in the center and to the interior of the end longitudinal extension plates. Stainless steel eyebolts (visible in Figure 33) were attached to each rib to protrude from the finished foil as the foil stay attachments. The hydrofoil was designed to be free flooding to facilitate neutral buoyancy, and, at one-half-scale, have an overall span of 165 inches with a chord of 10 inches

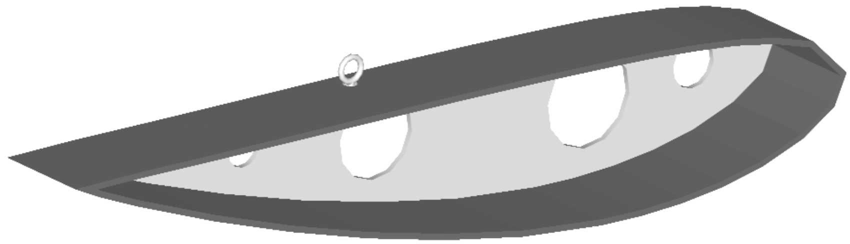

Figure 32. A half-scale hydrofoil rib.

The ribs (see Figure 33) were the primary structural component defining the hydrofoil's cross-section shape. The hydrofoil chord was 9⅞ inches long. Construction for each rib was entirely of ⅛-inch thick aluminum. The "flange" plate for the rib is ⅛-inch at both full and half-scale due to the manufacturing necessity of rolling the plate around the rib's web. The holes were passageways for the two aluminum pipes.

The structural design concept is illustrated in Figure 32 and half-scale dimensions (in inches) are provided in Figure 34. The half-scale pipes were to be ¾-inch schedule 40 and ½-inch schedule 40, respectively. At full-scale, the pipe designations are 2-inch schedule 40 and 1-¼-inch schedule 40, respectively. Though half-scale diameters are not exactly half the full-scale diameters, these choices represent the best Froude-scaled compromise with respect to both diameter and weight considerations. Each of the four foil end flanges, were cut from 3/8-inch aluminum plate and welded to the ends of both foil halves. The 3/8-inch plate is specified for full-scale, but was not scaled down due to concerns of possible warping during welding.

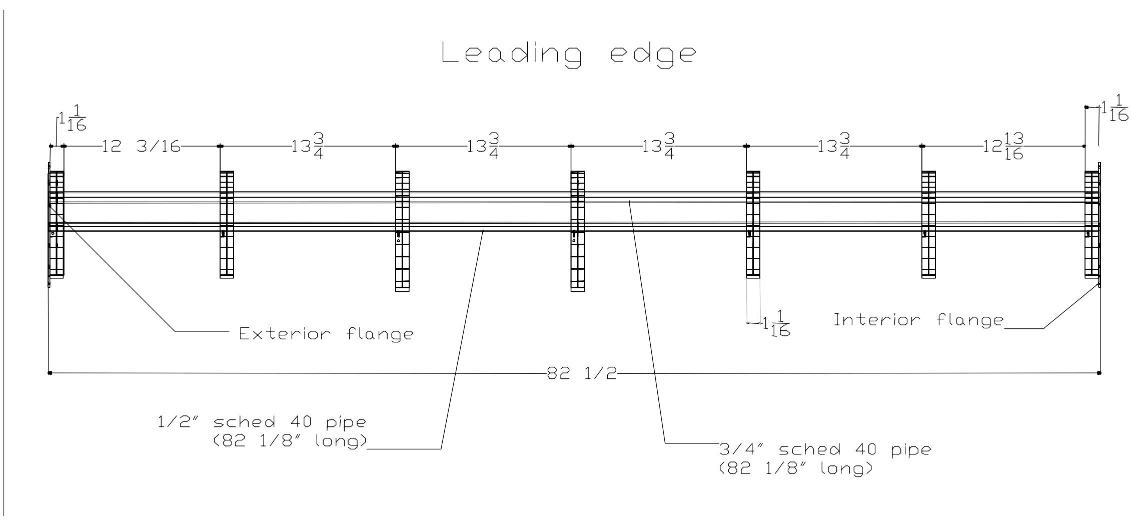

Figure 33. Structural design (top view) of half the half-scale hydrofoil.

Each rib was tack-welded to the pipe at the intervals prescribed (half-scale), which correspond to the distances between the intermediate longitudinals on the model's forward assembly (See Figure 34). This is so a hydrofoil stay may be run between the eyebolt on the hydrofoil and the intermediate longitudinal. The pipes were fully welded to each end flange, so that they were watertight. To assemble the hydrofoil, the interior flanges were bolted together, and the exterior flanges were bolted to the hydrofoil attachment plates, extending down from the end longitudinals (see Figures 35 and 36). The half-scale skin was fabricated of 0.08-inch aluminum sheet, corresponding to 0.16-inch aluminum sheet full scale, and attached to the ribs by pop-riveting it to the flange on each rib.

Figure 34. Front view of the half-scale Hydrofoil/Fast-Sweep model.

Figure 35. Front view of the half-scale Hydrofoil/Fast-Sweep model to show hydrofoil stays and hydrofoil attachment to the hydrofoil extension plate.

Cable stays attached the hydrofoil to the submergence plane as seen in Figures 35 and 36. Each stay was strong enough to transmit the load from the hydrofoil to the intermediate longitudinal, yet small enough to minimally interrupt the oil-contaminated water flow around it. There were 12 stays, one for each intermediate longitudinal, and two for the center intermediate longitudinal to lead to each half of the hydrofoil. The full-scale stays were designed as $^{3}/_{16}$-inch steel wire rope, but the half-scale stays were designed as ⅛" steel wire rope. This diameter was chosen as it is the smallest available to be easily swaged into an eye and for its 2100-pound working strength (McMaster-Carr, 1998).

The stays were swaged directly to the eyebolt extending from the intermediate longitudinal. The lower end of the stay was swaged into an eye and attached using a quick-release clip. With this technique, there would be no connecting link to further disrupt flow near the submergence plane, and the stays could be disconnected from the hydrofoil and folded into the forward assembly for transport.

The main purpose of the hydrofoil is to provide a downward force, but it also serves as a compression member. The half-scale hydrofoil's maximum compressive load was estimated to be 2/3 of the maximum chain tension force of 1,800 pounds equal to 1200 pounds. A conservative buckling analysis of the hydrofoil was completed using Euler buckling analysis (see Dugan, 2000), and it was found the hydrofoil could support 3,695 pounds of compressive force. This analysis modeled the hydrofoil as a 164.5-inch simply supported column and the skin as the structural member resisting the compression. The interior pipes were not included in the analysis, as their contribution to the area moment of inertia was an order of magnitude less than the skin, and was considered negligible.

The total in-water weight of the hydrofoil was determined to be 14.7 pounds. This assumed the two pipes to be airtight and the hydrofoil to be flooded. The impact of this weight on the forward assembly, supported by at least 2500 pounds of reserve buoyancy was deemed to be negligible.

Miscellaneous Components

The forward assembly wracking cables for the half-scale model were made from galvanized steel wire rope having a 7000 pound working capacity. Each of two cables spanned diagonally across the top of the forward assembly, and each end was outfitted with thimbles and crimps. Turnbuckles were included to remove sag. The required strength of these cables was not known exactly, but stresses were believed to be well below their operational limits.

A lift bridle was fabricated from 4 lengths of the same wire rope as that used for the wracking cables. Each length connected to a lift point on the corners of the forward assembly. All legs were linked to a common, central lift-ring above the center of the forward assembly. A static analysis confirmed that the bridle geometry and wire rope strength would be sufficient to allow an adequate overhead factor of safety of 5 when lifting the entire model weight of 500 pounds.

The model tow bridles were fabricated from the same wire rope used for the lift bridle and wracking cable. Load cell tests with the 1/5-scale model indicated full size tow forces of 10,400 pounds per end longitudinal. This would scale to nearly 1,300 pounds for the half-scale model. The 1,300 pounds was assumed to be evenly distributed amongst the four tow connections on each end longitudinal so the force on each cable would be much less than the working load of 7,000 pounds. Thimbles, crimps and shackles were used to complete the bridles.

Containment Boom

The inflatable half-scale Fast Sweep boom was fabricated much like the actual full-scale boom. It was made from the same urethane fabric and contained 3 air bladders per leg, which like the full-scale system, allowed articulation. Each of the three lengths of boom attached to each other with full-scale ASTM connector lengths cut in half (see Figure 31). An extra pair of these connectors was welded to the aft end of each end longitudinal for boom connection to the forward assembly.

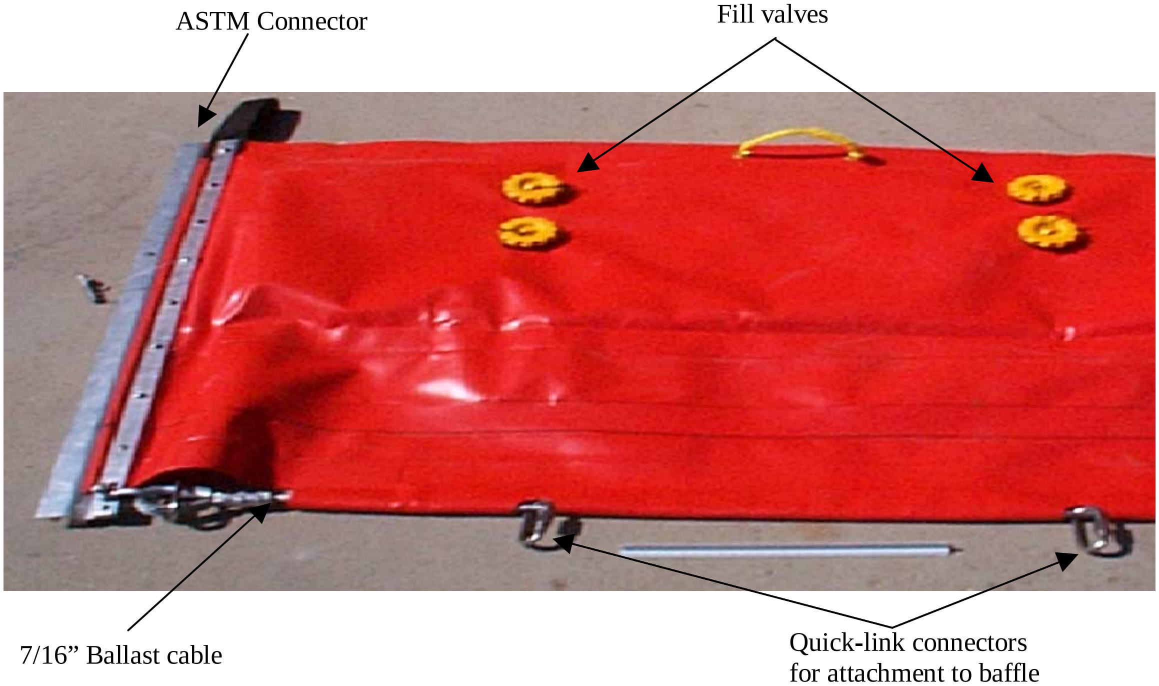

Figure 36. The half-scale Fast Sweep boom.

There were two substitutions in the fabrication of the half-scale boom. One was the ballast chain of the full-scale system. Instead of the ½ inch grade 80 galvanized ballast chain inserted along the boom length in the boom pocket, the half-scale boom used a 7/16" galvanized steel cable. At 0.31 lbs. per foot, this closely approximated the Froude-scaled weight of the full size chain of 2.61 lbs. per foot. The other substitution was the use of 3 air bladders per leg as opposed to 6 per leg on the full-scale boom. Since the chief cost component of the fabrication was the air bladders, the number of bladders was reduced while still maintaining a desired level of articulation.

System Assembly

The sequence of steps was critical to efficient and easy assembly. The first task was to connect the baffle chain prior to inserting and assembling the compression members and hydrofoil. Since the end longitudinals could be brought as close to each other as necessary, the chain was easily connected. Next, the aft compression member was inserted, followed by the forward member and the hydrofoil. Despite this preferred method of assembly, the compression tubes once inserted into their end longitudinal bases were difficult to align and sleeve with the center assembly since their alignment worked against the tension of the baffle chain. Likewise the hydrofoil was difficult to assemble within the width between the two end longitudinals. These problems were mitigated to a large extent by using waterproof marine grease. This grease when pasted onto the compression tubes and the interior of the hydrofoil extension plates reduced the burring and binding inherent in aluminum-to-aluminum interfaces. It was noted that it took 3 people just under 2 hours to erect the entire system (including hard connecting the submergence plan and baffle). This was encouraging, as a logistical mandate had been the full-scale system assembly by 4 to 6 people in 90 minutes.

Float Test

Following assembly, the entire system was placed into UNH's engineering tank. The system was observed visually for trim and draft characteristics. Overall, the system floated close to the design configuration, and the flexible aft portion took shape as intended. Slightly more freeboard was observed at the forward end due to the forward float. This was attributed in part, however, to under-inflation of the side floats and boom. The forward float did not span between the end longitudinals. A 5 – 6 inch gap existed on each side of the float between its end and the end longitudinal. It was speculated that this gap might cause problems by letting water into the device, and it was decided to observe such a tendency during testing.

VII. HALF-SCALE MODEL TESTING

Purpose

Testing of the half-scale Hydrofoil/Fast-Sweep model took place at the Ohmsett test facility. Testing days were Tuesday, 30 May 2000 through Friday, 02 June 2000, inclusive. The goals of the tests were:

- Visually observe the hydrofoil's ability to restrict the model's submergence plane rise at various tow speeds, with and without waves
- Quantitatively measure the model's ability to collect oil at various speeds, both with and without waves
- Visually observe the sea-keeping ability of the model
- Verify the structural design of the model
- Visually observe the hydrofoil's impact on oil-contaminated water flow under the submergence plane
- Quantitatively evaluate the impact of moving the hydrofoil from the upper to lower position on the model's oil retention
- Visually observe the impact of moving the hydrofoil from the upper to lower position on the model's sea-keeping ability
- Quantitatively evaluate the effect of changing the exit area on oil retention
- Quantitatively evaluate tow forces on the model.

Test Environment

During the testing period, the wind was negligible. There was no rain, and the tank's water temperature ranged from 64 to 67 degrees Fahrenheit. The Ohmsett tow tank is shown in Figure 37.

Length	203 meters
Width	20 meters
Depth	3.4 meters
Maximum Tow Speed	6.5 knots
Sundex 790 oil:	
Specific Gravity	0.96
Viscosity (v)	10,000 cSt

Figure 37. Ohmsett tow tank and dimensions.

The forward end of the tank contains a flap-type wave maker, and the aft end supports the "beach" designed to absorb the wave energy. The beach can be raised or lowered resulting in regular waves, similar to offshore waves, or reflected waves, similar to harbor chop. The tank has three tow bridges one auxiliary and one fore and one aft of the moving test area.

The forward bridge towed the model, released the oil onto the water, and housed the control booth. After tests were complete, oil was pumped from the model into collection containers, on the aft bridge, from which oil samples were analyzed. The aft bridge was also used to secure the aft portion of the model for return tows.

Test Protocol and Schedule

The half-scale Hydrofoil/Fast Sweep model was assembled on the splashdeck of Ohmsett in about two hours. After the system was lowered into the tank, its bridle lines were secured to the forward tow bridge by two six-foot towlines. These lines were galvanized steel wire rope with a 7,000-pound working load. Care was taken to ensure the towlines were at the waterline and parallel to the waterline. At the start of each test, the forward bridge was slowly ramped up to the final steady-state speed. The time it took to achieve steady-state speed varied with the magnitude of the test speed. Load cells attached in series to the tow lines recorded tow forces applied to each end longitudinal at one-second intervals. Sundex 790 oil was released at approximately 25 gallons per minute after the model reached the final tow speed. Sundex 790 has a specific gravity of 0.96 and a kinematic viscosity of 10,000 centistokes (cSt). If the tow included waves, then the waves were allowed to fully develop before the tow began. Following the tow, the captured oil in the model was pushed to the model's aft end using fire hoses from the forward tow carriage, and pumped into the collection tanks on the aft tow carriage. The model was then slowly towed backward to prepare for the next test. Several speed tests (without oil) were also done both with and without waves.

The apex region (aft-most portion) of the containment region was opened up by disconnecting the horizontal baffle from the Fast Sweep skirt across the stern for some tests to investigate the effects of increasing exit area. This is termed the “open apex” configuration as opposed to the normal “closed apex” configuration in which the Fast Sweep skirt at the stern is continuously attached to the horizontal baffle.

Each tow was videotaped with an overhead camera from the front tow bridge, an underwater camera from the front tow bridge, and an underwater camera from the aft bridge. In addition, portable video and digital cameras were used. A condensed test schedule, listed by date and type of tow, is presented in Table 1.

Table 1. Half-scale Hydrofoil/Fast-Sweep model test schedule at Ohmsett.

Date	Test speeds, knots	Remarks
30-May-00	0.71, 1.4, 2.1, 2.8, 3.5	Preliminary sea-keeping tests, no waves, no oil
31-May-00	0.71, 1.4, 2.1, 2.8, 3.5, 4.2, 5	Speed and oil tests, closed and open apex
1-Jun-00	2.8, 3.5, 4.2, 5, 5.5, 6, 6.5	Speed and oil tests, waves included, closed and open apex
2-Jun-00	2.8, 3.5, 4.2, 6	Hydrofoil in lower position for all tests, closed apex Speed and oil tests, waves included

A total of 52 tests were conducted. Eleven of the tests were preliminary, and as such, were not assigned an official test number. Oil was placed into the tank for 28 of the tests. The remaining 24 tests involved examinations of the device’s speed range, bow attitude, sea-keeping characteristics, and the effect of the hydrofoil. The testing schedule was flexible, dynamic and subject to immediate adjustment as the effects of influences such as the hydrofoil and the boom were being observed. Because oil retention experiments involve multiple and conflicting scaling considerations, actual speeds are given. For runs without oil, Froude scaling is appropriate and the corresponding full-scale speed is found by multiplying model speed by $2^{1/2}$.

Preliminary Tests

The first set of runs was conducted without oil in both calm water and with waves. The purpose of these preliminary tests was to obtain observations of the system’s attitude (bow angle) and sea-keeping characteristics. First, the model was towed at 0.71, 1.4, 2.1, 2.8 and 3.5 knots corresponding to 1, 2, 3, 4, and 5 knots full-scale. Throughout the speed range, the forward assembly rode nearly level, had little change in vertical position and appeared very steady and stable. At the low speeds, the gap between the end longitudinals and the forward float caused water to flow into the forward assembly. Also at low speeds, the boom necked in from both sides. This was believed to be due to inadequate flow through the device due to the low speeds. The necking was observed to lessen at 2.8 knots and higher, however, at the higher speeds the leakage between the forward float and the end longitudinals became jet-like. Leakage also worked its way, around the submergence plane/angle attachment, from the high-pressure (lower) side of the submergence plane into the containment region. At 3.5 knots, there appeared to some observers to be a slight bow up attitude. However, it was minimal and verified that the hydrofoil was working.

Next, the device was tested at 0.71, 1.4, and 2.1 knots in waves having a wavelength of 19.5 feet, a wave steepness of 1 /15, and a period of 2 seconds. This corresponded to a wavelength equal to half the length of the system, and was deemed an extreme condition. The model did not exhibit necking or instability. The wave conditions seemed to mitigate the "leaking" of water into the forward assembly, as there was minimal leaking between the end longitudinals and submergence plane. The intermediate longitudinals were seen to "slap" the submergence plane resulting in a "cupping" of the submergence plane beneath. It was thought that if the intermediate longitudinals were better secured at their upper forward ends, (i.e. pocketed), that this would not occur.

Main Test Program

The 41 experimental runs in the test program were conducted with and without oil in both calm and wave conditions. Each run was documented by Ohmsett and assigned a run number. The test program with results is summarized in Table 2, while retention results in Table 3 are reorganized by test type. The first tests were run at 0.71, 1.4, 2.1, 2.8, 3.5, and 4.2 knots with the hydrofoil in the upper (normal) position and the apex closed (normal), as shown in Figure 38.

Table 2. Ohmsett test program summary.

	31-May-00										
Test #	1	2	3	4	5	6	7	8	9	10	11
Tow Speed	0.71	1.42	2.1	2.8	3.5	4.2	4.2	5	4.5	4.2	4.2
Vol Dist.(gal)	93	126	140	118	105	n/a	70	n/a	n/a	n/a	n/a
Waves	no	no	no	no	no	no	no	no	no	no	no
Apex Congiguration	closed	closed	closed	closed	closed	closed	closed	open	open	open	open
Foil Position	upper	upper	upper	upper	upper	upper	upper	upper	upper	upper	upper
Recov Vol oil (gal)	51.71	125.94	133.86	91.89	53.08	n/a	31.07	n/a	n/a	n/a	n/a
Retention	55.6	99.95	95.62	77.87	50.55	n/a	44.38	n/a	n/a	n/a	n/a

	31-May-00				1-Jun-00						
Test #	12	13	14	15	16	17	18	19	20	21	22
Tow Speed	5	5	4.2	3.5	2.8	4.2	3.5	2.8	2.8	3.5	5.5-6
Vol Dist.(gal)	n/a	92	106	124	113	102	121	121	122	124	n/a
Waves	no	no	no	no	no	no	no	no	yes	yes	no
Apex Congiguration	open	open	open	open	open	open	open	open	open	open	open
Foil Position	upper	upper	upper	upper	upper	upper	upper	upper	upper	upper	upper
Recov Vol oil (gal)	n/a	14.03	32.16	49.77	63.52	31.93	46.78	99.37	68.94	19.76	n/a
Retention	n/a	15.24	30.34	40.14	56.21	31.3	38.66	82.12	56.51	15.94	n/a

	1-Jun-00										
Test #	23	24	25	26	27	28	29	30	31	32	33
Tow Speed	6	6.5	6	2.8	3.5	4.2	2.8	3.5	5	2.8	3.5
Vol Dist.(gal)	n/a	n/a	n/a	117	116	120	140	163	n/a	126	121
Waves	no	no	no	no	no	no	yes	yes	no	no	no
Apex Congiguration	open	open	open	closed	closed	closed	closed	closed	closed	closed	closed
Foil Position	upper	upper	upper	upper	upper	upper	upper	upper	upper	lower	lower
Recov Vol oil (gal)	n/a	n/a	n/a	55.17	46.2	45.37	75.29	30.17	n/a	63.52	52.25
Retention	n/a	n/a	n/a	47.16	39.83	37.81	53.78	18.51	n/a	50.41	43.18

		2-Jun-00						
Test #	34	35	36	37	38	39	40	41
Tow Speed	4.2	2.8	3.5	4.2	2.8	3.5	5	6
Vol Dist.(gal)	115	126	118	84	124	118	n/a	n/a
Waves	no	no	no	no	yes	yes	no	no
Apex Congiguration	closed	closed	closed	closed	closed	closed	closed	closed
Foil Position	lower	lower	lower	lower	lower	lower	lower	lower
Recov Vol oil (gal)	46.45	68.74	52.3	29.99	58.89	34.59	n/a	n/a
Retention	40.39	54.55	44.32	35.7	47.49	29.31	n/a	n/a

Table 3. Summary of averaged oil collection percentages.

Speed (knots)	Closed Apex, Upper pos., No Waves	Open Apex, Upper pos., No Waves	Closed Apex, Lower pos., No Waves	Closed Apex, Upper pos., Waves	Open Apex, Upper pos., Waves	Closed Apex, Lower pos., Waves	Average, No Waves	Average, Waves	Average, All
0.71	55.6	N/A	N/A	N/A	N/A	N/A	55.6	N/A	55.6
1.40	99.95	N/A	N/A	N/A	N/A	N/A	99.95	N/A	99.95
2.10	95.62	N/A	N/A	N/A	N/A	N/A	95.62	N/A	95.62
2.80	62.02	69.17	52.98	53.78	56.51	47.49	61.39	52.59	56.99
3.50	45.19	39.4	43.75	18.51	15.94	23.91	42.78	19.45	31.12
4.20	37.81	30.82	38.05	N/A	N/A	N/A	35.56	N/A	35.56

**Figure 38. Half-scale model in Ohmsett's tow tank,
2.8 knots, closed apex**.

At 0.71 knots, oil was observed to "pile" up in front of the forward float. It appeared that the incident velocity drove the oil down the submergence plane very slowly and the oil did not reach the gap until nearly the end of the test. Oil did not get past the system and the wake was clean. Despite not being in steady state for most of the run, 56 percent of the oil was recovered from the containment region. Nearly all the remaining oil was still in front of the submergence plane. At 1.4 knots, necking of the boom reduced the opportunity for oil missing the gap to enter the containment region through its exit holes. Some of the oil was entrained past the gap and was lost from the containment region under the boom sides. More water appeared to be exiting the device than entering through the gap thus creating a continuity problem. This was thought to be due to water jetting in from between the forward float and the end longitudinals. These deficiencies were minor, however, since nearly 100 percent of the oil was retained. At 2.1 and 2.8 knots there was little necking of the boom, resulting in retentions of 96 percent and 78 percent, respectively. At 3.5 knots there was no necking, but significant oil was missing the gap and there was entrainment out the exit holes resulting in retention of 51 percent. The 4.2-knot test without oil showed spillover at the apex was imminent. The 4.2-knot test with oil showed some intermittent oil spillover in addition to the losses seen at 3.5 knots. Nearly half of the oil (44 percent) was collected.

Apex Modifications

With limited testing time available, it was decided to concentrate the remaining tests on the high-end speeds. Since at speeds over 4.2 knots oil went over the top of the boom, the apex was opened to induce the flow to remain under the Fast Sweep stern. This approach had been demonstrated previously using the 1/5 scale model in the UNH tow tank.

The apex was first modified by extending the connection between the aft end of the baffle and the boom skirt at the squared-off stern between them. This was done by inserting 2 extra quick-links into the baffle-boom stern connection. This resulted in 7 ½ additional inches of opening along the aft edge of the apex. At 4.2 knots, underwater video showed the baffle flapping like a flag and pulling the apex under constantly. It was held under continuously over 4.5 knots. The apex pull-under and baffle flag-like action would not have retained oil in the collection region so the apex was opened entirely by disconnecting the quick links between the boom stern and the baffle. Under this condition, the apex was stable on the surface at 4.2 knots. Again at 5 knots without oil, it was stable.

Later, the boom stern / aft baffle end interface was opened further by disconnecting all quick-links which previously joined the two. Though there was no spillover, at 5 knots, just over 15 percent of the oil was collected. Retention improved to 30 percent (see Figure 39) at 4.2 knots. Underwater cameras showed that the bow turbulence appeared to break the slick into a cloud and disperse it. At 3.5 knots, the underwater video showed the baffle trailing instead of flapping and oil retention was 40 percent. Oil retention increased to 56 percent at 2.8 knots with the apex open.

Figure 39. Half-scale model in Ohmsett's tow tank, 4.2 knots, open apex.

These collections were somewhat below those of the closed apex configuration (see Table 3). It was thought that perhaps the open apex configuration might be better suited for high-speed survivability travel, and the closed apex configuration better for oil collection. It was desired to conduct replicate tests for the open apex configuration. Tests at 2.8, 3.5 and 4.2 knots gave collection figures of 82 percent, 38.66 percent, and 31.30 percent, respectively.

Following this, the apex was closed as originally tested. Replicate tests were conducted with oil. Collection percentages were somewhat close to the earlier tests in the closed-apex configuration at the same speeds at 47.16 percent, 39.83 percent, and 37.81 percent for 2.8, 3.5 and 4.2 knots, respectively.

Oil Collection in Waves

Oil collection ability was tested in waves with wavelengths of 29 feet (equal to system length) corresponding to a period of 3 seconds in the 8-foot deep tank. Wave height was 1.5 feet—the maximum possible due to stroke length limitations at Ohmsett. Though shorter waves and / or harbor chop might have been used, it was noted that in the field, chop would mix oil vertically in the water column so mounting any surface skimming effort would be questionable.

Testing in waves was initially conducted with the apex in the open configuration. At 2.8 and 3.5 knots, collection of the oil was 56.51 percent and 15.94 percent respectively. There appeared to be good system contouring and no forward assembly dynamical instabilities. Oil was seen to escape out through the open apex with the passing of wave crests. There was no necking of the boom.

Later, 2.8 and 3.5 knot testing was conducted in the closed configuration (see Figure 40). At 2.8 knots with a closed apex, the apex did not submerge, however, there was oil slopping over the boom (see Figure 41). Another loss mechanism was vertical oscillation of the horizontal baffle. It was postulated that the Fast Sweep boom skirt would benefit from a ballast chain to prevent this. At 3.5 knots, the wave crests again contributed to oil loss over the stern boom. Oil collection during the 2.8 and 3.5 knots runs were 53.78 percent and 18.51 percent respectively.

Figure 40. Half-scale model in Ohmsett's tow tank, 3.5 knots, waves, closed apex.

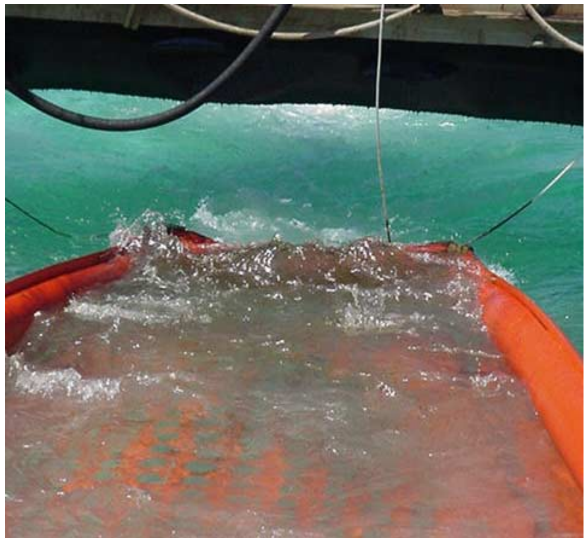

Figure 41. Spillover at the apex as wave crest passes, 2.8 knots, closed apex.

System Survivability Tests

As the full-scale device was specified to withstand up to 10 knots, the half-scale model was tested up to the limit of tow carriage speed. At 5.0 knots without oil, the aft compression member was seen to bow slightly due to baffle chain tension. To see if the model could indeed withstand the extreme hydrodynamic forces at top-end speeds, it was run down the tank without oil at 6 knots, (8.45 knots full-scale). At this speed, there was a slight downward attitude of the bow due to the hydrofoil. The aft end of the submergence plane was draining. The system was then run at the maximum speed of the tow carriage, 6.5 knots, corresponding to 9.15 knots full-scale. Again the bow was stable; the aft end of the submergence plane was drained. Additionally, the water inside the forward assembly was highly foamed, as shown in Figure 42.

Figure 42. Half-scale model in Ohmsett's tow tank, 6.5 knots (9.2 knots full-scale), open apex.

Hydrofoil Position Variability

During the last series of tests, the model operated in the closed apex configuration with the hydrofoil in the lower position, 6 inches below the previous location. Oil collection sensitivity to hydrofoil placement, in both the calm and wave states, was evaluated. Tests at 2.8, 3.5, and 4.2 knots including replicates were conducted. Visible observations were very similar to previous oil tests at the same speeds. Oil collection averaged 52.98 percent at 2.8 knots, 43.75 percent for 3.5 knots, and 38.05 percent for 4.2 knots. These figures were very close to the collection percentages of the corresponding speeds with the upper hydrofoil position. Next, 2.8 and 3.5 knot collection was tested in waves. Percentages were 47.49 percent and 29.91 percent, somewhat close to the collection figures corresponding to the same speeds in the upper hydrofoil position. Lastly, the model was tested at the higher speeds. At 5 and 6 knots, the hydrofoil held the system down well. An upward bending of the aft compression member was again noticed. At 6 knots, the aft barrier was continuously under water. During these tests, the model sustained minor damage in the form of some loosened fasteners and two un-linked hydrofoil stays. The hydrofoil flange sustained bending between bolts at the mid-foil junction.

The hydrofoil in the upper position did not appear to interfere with the water and oil-contaminated water flow into the mode (as seen from underwater video). When the hydrofoil was examined following testing, it had relatively little oil on it. This indicated very little bodily interference with the oil-contaminated water flow. On the other hand, the underside of the submergence plane, located 15 inches above the hydrofoil, was covered with oil. The position of the hydrofoil did not seem to affect the oil capture performance.

The model's sea-keeping ability also did not seem to be affected by the position of the hydrofoil. There was some concern that the longer moment arm created by lowering the foil would give the forward assembly a bow-down attitude. No bow-down attitude, nor any other attitude change, was evident during any of the tests with the hydrofoil in the lower position.

Oil Collection Processes

Though hydrofoil performance and model sea-keeping was excellent at all speeds and in all speed/wave combinations, oil collection had been less than expected (see Tables 2 and 3 and Figure 43). Problems in oil collection percentages stemmed mainly from bow-induced turbulent mixing and the Fast Sweep / baffle configuration. The square corners of the Fast Sweep stern accumulated vorticity, which then induced draining at the exit area. Insufficient ballasting of the Fast Sweep skirt allowed the horizontal baffle to ride up in waves forcing exit area losses. At lower speeds of 2.1 and 2.8 knots, boom necking and baffle contraction occurred, resulting in loss of oil under the boom skirt and baffle. The results of the experiments with the two apex configurations (open and closed) indicated clear results. The closed apex configuration produced oil collection percentages higher than those associated with the open apex configuration so this is the preferred configuration for oil collection. However, both variations had very similar collection percentages in waves. It is believed that an open apex reduces the drag forces will this causing less stress on the boom baffle system when being towed at the top-end survivability speeds.

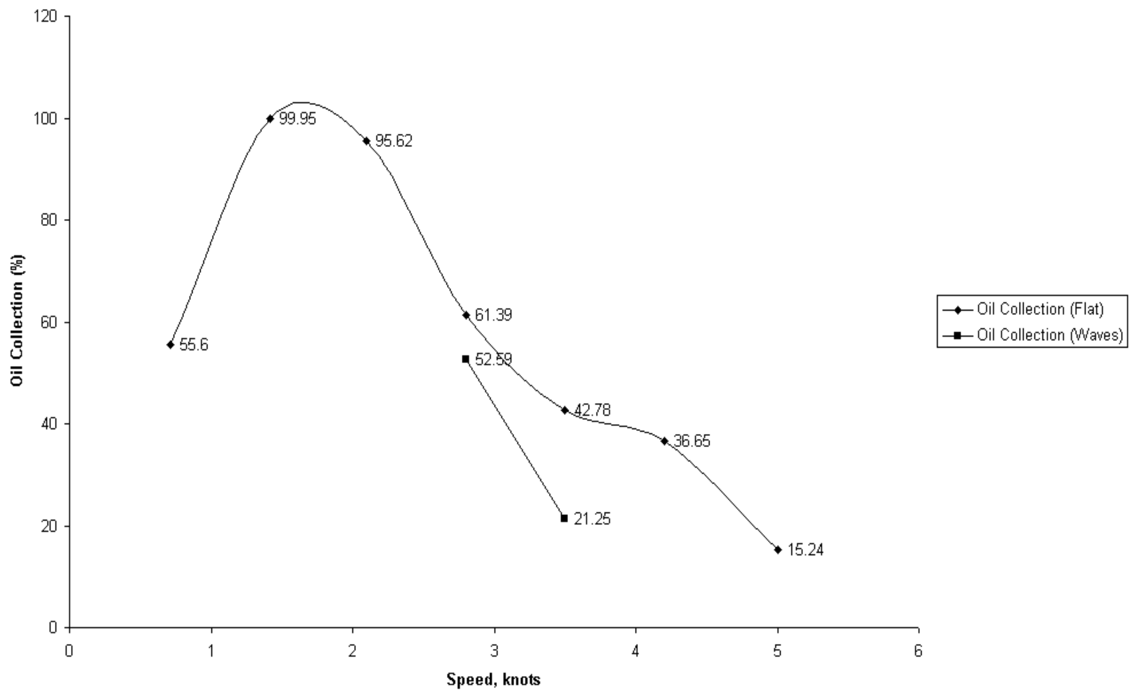

Figure 43. Plot of the half-scale model's oil collection at Ohmsett.

Figure 43 shows the drop-off in retention percentage with increase in model speed. Results were not scaled up to full-size because the general position of the scientific community is that there are too many conflicting scaling considerations to do this correctly. Expected trends, however, may be described based on knowledge of submergence plane oil retention mechanisms. Consider the comparison of model retention and full-scale retention at the same speed for the same oil. Since oil droplet rise velocity is the same and the full-scale gap is larger, there would be more time (greater horizontal distance) for oil to rise up through the gap as well as less vertical movement required (deeper bite). The higher gap capture rate would yield better retention at full-scale. Following this reasoning, full-scale oil retention at a convenient operating speed of 3 knots would be at least the demonstrated minimum of 60 percent and probably much better.

Load-Cell Data

In all main test program experiments, load cells attached to each end longitudinal recorded tow forces. Loads were recorded by the load-cells in real time at one-second intervals from the start of tow carriage movement, until it stopped at the end of the test. Load recordings increased as the tow carriage accelerated to steady-state speed. Once steady-state speed was achieved, the load-cell recordings leveled off to nearly constant values. The load-cell values for the steady-state portion of the data were averaged together for each test speed. Average tow force per end longitudinal values for replicate tests were then averaged together. These tow force per end longitudinal average values were then plotted against speed. As seen in Table 4 and Figure 44, the maximum tow forces were recorded at speeds of 6.0 to 6.5 knots corresponding closely to the scaled survivability speed. Surprisingly, the differences in half-scale tow force magnitude between 2.8 and 3.5 knots in waves and the same speeds without waves was small (34.82 pounds, and 69 pounds respectively). As such, the half-scale tow force values shown in Figure 44 include all tows, including those with waves. The tow and wave tests conducted at 1/5 scale earlier were scaled up to full size and compared with the half-scale results scaled up to full size in Figure 44. A close correlation between the 1/5 scale forecast and half-scale scale-up can be seen.

Table 4. Summary of half-scale model end longitudinal tow forces and equivalent full-scale tow forces per end longitudinal.

Speed (knots)	Speed (knots full-scale)	Half-scale Avg. Tow Force per end long. (lbs)	Equiv. Full Scale Avg. Tow Force per end long. (lbs)
0.71	1	19.05	152.4
1.42	2	73.24	585.92
2.1	2.96	151.72	1213.76
2.8	3.94	254.14	2033.12
3.5	4.93	372.44	2979.52
4.2	5.92	485.68	3885.44
4.5	6.34	553.17	4425.36
5	7.04	756.75	6054
5.5	7.75	790.67	6325.36
6	8.45	999.82	7998.56

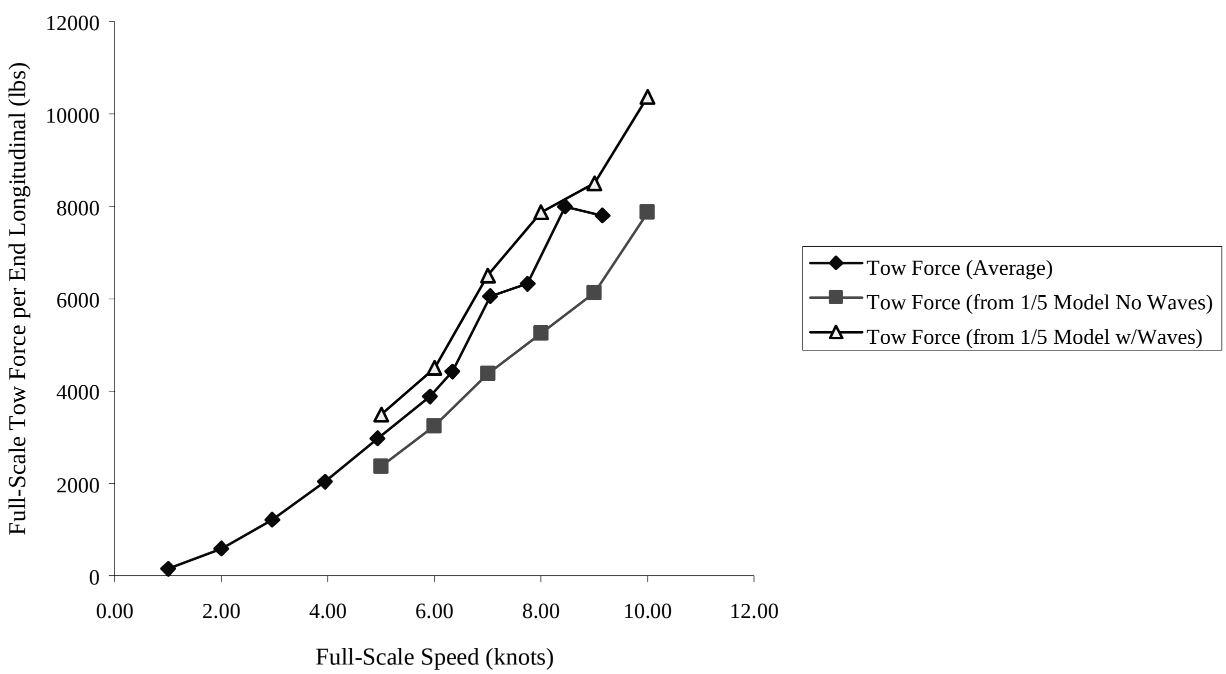

Figure 44. Comparison of 1/5 scale model and half-scale model tow test data.

A trend line was established from the half-scale total drag force data as indicated in Figure 45. Although maximum tow force scaled to full-scale was 7804 pounds at the highest test speed of the half-scale model (6.5 knots, 9.15 knots full-scale), this trend line indicated a tow force of nearly 10,400 pounds per end longitudinal at 10 knots full-scale. This value of 10,400 pounds per end longitudinal was used for the strength analysis of the system discussed in the next section.

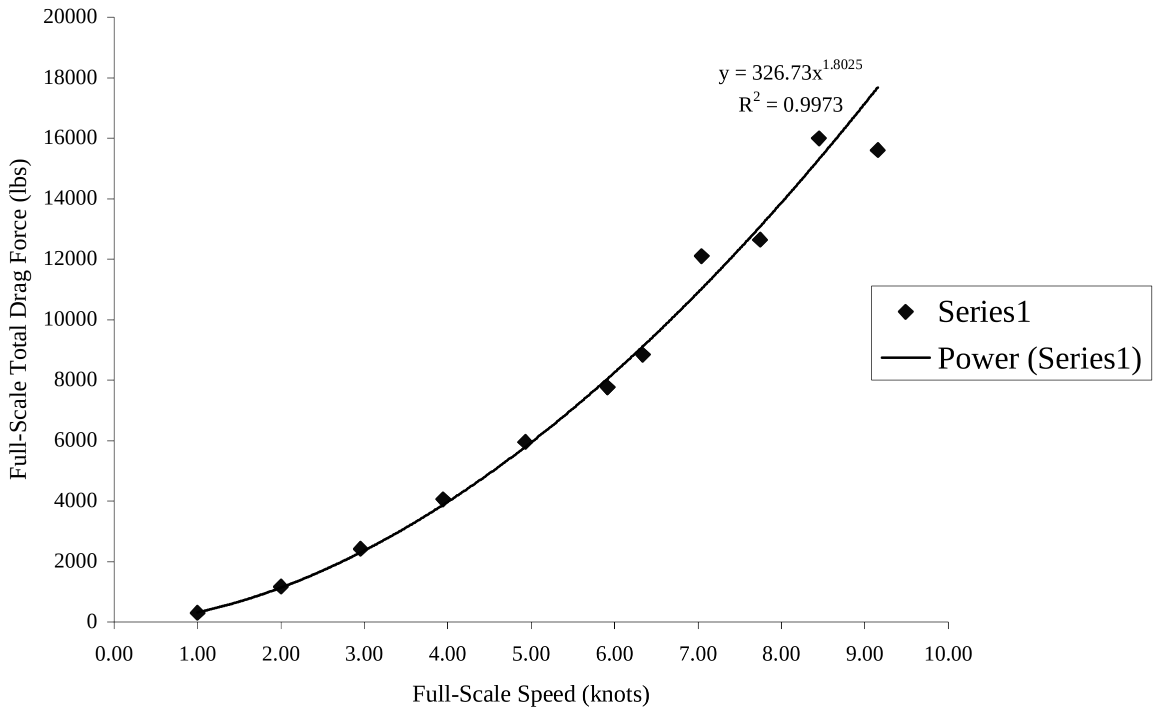

Figure 45. Full-scale tow force vs. speed trend line derived from half-scale test data.

Determination of Coefficient of Drag

Figure 45 shows drag force of the full-scale system as a function of speed. The full-scale drag force values were determined by multiplying the recorded tow force per half-scale model end longitudinal by two and scaling to full-size.

The trend line is a very close fit to the actual data. As expected, the data shows a velocity-dependent tow force, which closely approximates a squared function. The equation of the trend line is:

$$T = 326.73 * V^{1.8025} \tag{7}$$

where

T = Drag force (pounds of force, lb_f)

V = Speed of the system (ft/s).

Using the standard relationship for drag force, the standard quadratic mathematical form obtained from dimensional analysis,

$$F = \tfrac{1}{2}\rho C_D A V^2 \qquad (8)$$

Where

- F = force (lbf)
- ρ = fluid density (slugs/ft^3)
- C_D = drag coefficient
- A = the projected area (ft^2) of the object, and
- V = speed (ft/s).

Rounding up the power in equation (7) and equating the constant 326.73 to $\tfrac{1}{2}\rho C_D A$, where A is the projected area of the forward end of the full-scale device (75.94 ft^2), the drag coefficient is calculated to be 4.35.

Overall System Performance

Through the full speed range, zero to the maximum Ohmsett carriage speed of 6.5 knots (9.2 knots full-scale), the hydrofoil functioned very well in holding down the forward assembly. There was negligible vertical rise and no perceptible angle change. Wave motion did not induce any hidden instabilities. Since there was no performance benefit regarding the higher or lower foil mounting position, the upper position may be used, thereby reducing extension plate draft and weight. However, the overall shape of the forward end of the forward assembly, specifically the forward float, may have contributed to some bow wave turbulence problems. Turbulent mixing of oil causes some oil to miss the gap opening thereby reducing retention. Further research would be useful for determining an optimal shape and configuration. Another problem had been the inflow of water into the forward assembly area by means other than the gap. This can be corrected by suitable placement of sheet protection. Fast Sweep boom improvements include eliminating vorticity collection inside corners and skirt ballasting to hold down the horizontal baffle.

Oil collection had been good at minimal speeds necessary for vessel maneuvering and control. Nearly all oil was retained at 1.4 and 2.1 knots, while calm water average oil retention at 2.8 knots was 61 percent and wave motion degraded retention only slightly. At 3.5 knots, average oil retention without waves dropped to 43 percent, while wave motion degraded retention to 19.5 percent. As described in the oil collection processes section, there is reason to believe these percentages would increase at full-scale. The normal, closed apex more consistently retained oil compared to the open apex.

Moreover, the testing showed that the half-scale model, as an oil collection system in itself, would make an effective VOSS for a small vessel. With the model's lightweight and convenience in transport and storage, this small-scale, high-speed, oil skimming system is a viable option.

VIII. DESIGN OF FULL-SCALE PROTOTYPE SYSTEM

Design Development

The full-scale Hydrofoil / Fast Sweep design was based largely on the revised design implemented in the half-scale model described in Section 6. The design process consisted largely of scaling up the half-scale model to full-scale while correcting design problems that revealed themselves in the Ohmsett testing. No attempt, however, was made at this point to alter the forward assembly concept or change the Fast Sweep boom design since their usage was part of the design specification.

In Froude scaling the half-scale model to full size (see Appendix A), lengths will double; weights and forces will be 8 times larger, so stresses (force per area) will also double. Thus a strength analysis had to be done and design changes made to preserve a reasonable factor of safety. Components of the hard structure (end longitudinals, compression members, hydrofoil and chain) were subject to both a finite element analysis and strength of materials analysis. Redesign of highly stressed and / or weak components was carried out and new analyses were completed in an iterative cycle until adequate strength was achieved.

The final version is presented in this section where changes from the half-scale design are discussed. Drawings are provided in Appendix B. The strength analysis, as applied to the final version, is detailed in Appendix C.

Design Resources

For the full-scale design process, the AutoCAD 2000 drafting software was the chief tool used. Within its capabilities, the complete full-scale system was modeled electronically in three dimensions. Each salient component and sub-component was designed in a dedicated file. These many dedicated component files were then merged into one master system file. In the master file, each component was assigned a visual layer, which could be turned on or off to show views of any series of components. With its three-dimensional visual capabilities, solids modeling was employed to accurately determine component interactions. The program was also used to determine mass, volume, weight, area, moments of inertia, radii of gyration, and centroids. From the three-dimensional AutoCAD model, standard engineering drawings were produced by creating views of the model at various locations and distances while applying dimensions and notes.

End Longitudinals

The half-scale end longitudinals had been made from 1 ½-inch by 1 ½-inch by ⅛-inch square aluminum tubing. The full-scale end longitudinals would be made from 3 inch x 3 inch by ¼-inch tubing, making the Froude scaling mainly geometric. Several key modifications were made however.

First, each full-scale end longitudinal was modified to include a scroll-like piece of aluminum attached to its forward float shield (see Figure 46). This was incorporated to prevent water from entering the forward assembly by jetting between the end longitudinals and the ends of the forward float as observed during half-scale testing. The forward float shields were modified at the full-scale level to conform neatly to these scrolls. Cutouts in the shields would permit inflation of the forward float.

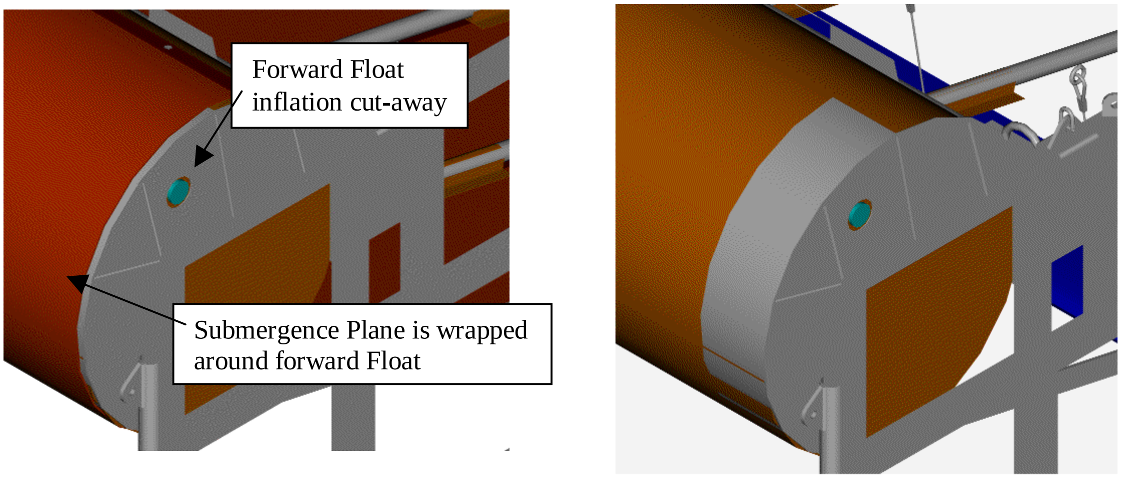

Figure 46. Full-scale revised forward float shield with incorporated scroll feature. At right, shown without submergence plane for clarity.

Another modification to the revised full-scale end longitudinal was increasing the height of its frame. The revised full-scale end longitudinal was made 4 inches taller to better accommodate the 32-inch Fast Sweep draft as actually built and deployed.

The hydrofoil extension plates were redesigned since there was no longer the need for two vertical hydrofoil positions. Oil collection results were very close for each position. It was, therefore, determined to use only one position for the full-scale design. The upper hydrofoil position was chosen, as this would allow a smaller extension plate. The revised extension plate was 15 inches smaller in height than the half-scale version scaled up. To maintain scaled horizontal distance to the gap, the angle of the forward end was modified from 45 to 34 degrees. The plating used for the hydrofoil mount on the full-scale extension plate is ¼-inch thick. This is the same thickness as the corresponding plate on the half-scale model. The ¼-inch plate was used at half-scale to ensure durability of the mount plate due to the numerous holes, which had to be drilled into it for two hydrofoil mounts. For ease of full-scale hydrofoil mounting, the interior framing of the extension plate was modified to provide better access to the flange boltholes. A slimmer, less obtrusive stiffening member replaced the mid-height box beam, which had been present on the half-scale extension plate.

The largest modification was the incorporation of stiffening members to provide adequate rigidity to the forward assembly as determined by an engineering analysis (discussed in Appendix C). The addition of these members, shown in Figure 47, prevent excessive out-of-plane bending induced into the end longitudinals as a result of baffle chain tension. The stiffeners consisted of variations of Tee-beam sections and gussets added to the end longitudinal in critical locations. One location was along the length of the bottom-most member of the end longitudinals. Another was along the length of the extension plates and in place of the mid-height box beams used at half-scale. The total weight of the additional stiffening members per end longitudinal was 160.72 pounds.

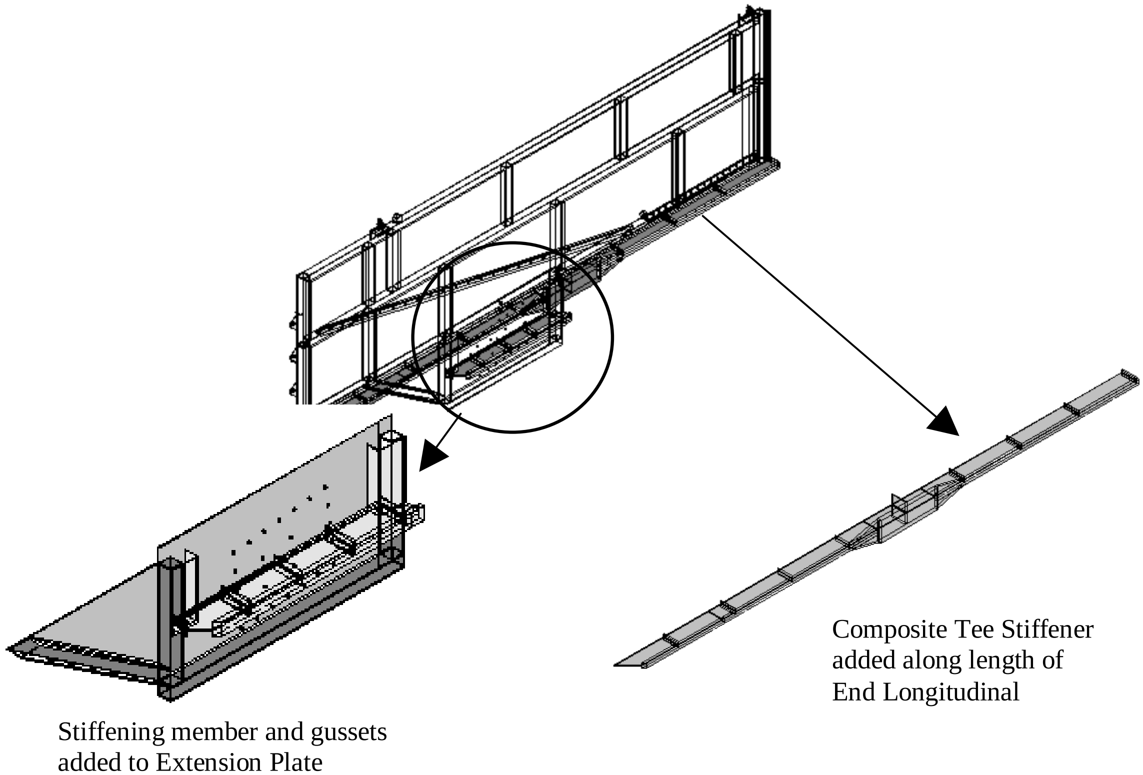

Figure 47. Structural modifications to end longitudinals (forward float shield not shown for clarity).

Intermediate Longitudinals

The full-scale intermediate longitudinals were designed from 2 ½-inch diameter schedule 40 aluminum pipes. The half-scale intermediate longitudinals scaled to full size would be heavier than their full-scale versions. However, as free-flooding members in a forward assembly with much reserve buoyancy, performance at the full-scale was not expected to deviate from that at half-scale. In addition, the top forward ends of the intermediate longitudinals would be securely fastened to the submergence plane with a fabric sleeve to reduce the “cupping” effect observed during half-scale testing.

Compression Members

The full-scale compression members are the same as for the half-scale model, but with heavier components. Analysis of the required strength at full-scale loading was conducted with a computer model using Euler buckling analysis using the half-scale model dimensions (discussed in Appendix C). The compression tubes were designed from 6 inch tubing, 5/16 inch thick. This tubing provided sufficient cross-sectional area moment of inertial to resist buckling under the entire chain load. Thus each of the two compression members alone have the capacity to withstand the chain tension load. The center assembly and compression bases were designed from 6-inch schedule 40 pipes, which, with an interior diameter of 6.065 inches, provide a snug but workable fit for the 6 inch outside diameter compression tubes. The compression members were also analyzed for shear of the locking pin rods. These pins were found to be sufficient in shear resistance. The forward float brackets welded to the forward compression member are designed at full-scale to be made of ⅛" thick aluminum sheet, just as those at half-scale. There are no major structural loads applied to these members; the ⅛-inch thickness was chosen to provide local strength during handling and storage.

Hydrofoil

The full-scale hydrofoil, like its half-scale predecessor, was designed entirely from aluminum. Design at full-scale is similar to construction at half-scale with the addition of strengthening longitudinals as seen in Figure 48. Details can be found in Dugan (2000), however, the major modifications were:

- Use of 0.16-inch thick aluminum sheet for the skin
- Main structure backbone to consist of one 2 inch and one 1 ¼ inch nominal diameter aluminum pipe, each running the length of the hydrofoil and connected to each interior rib
- Incorporation of aluminum bars running the length of the hydrofoil in its interior to increase the area moment of inertia about the weak axis to increase buckling resistance.

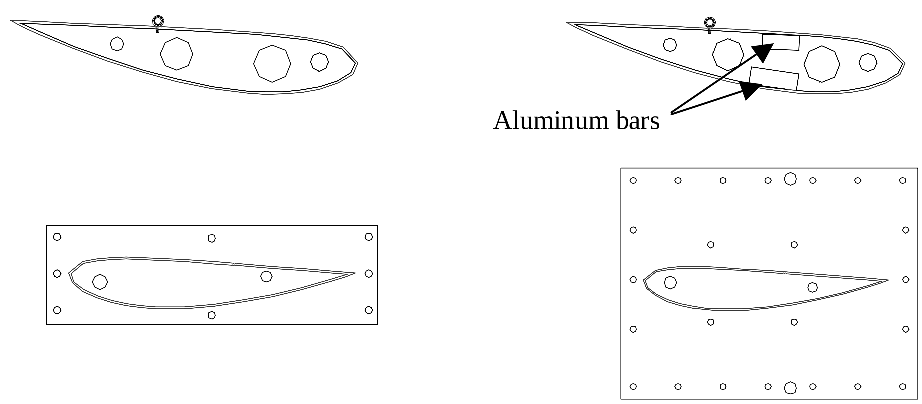

Figure 48. Full-scale design modifications to hydrofoil and flanges; at left, prior to modifications, and at right after modifications.

Miscellaneous

Froude scaling the half-scale baffle chain force of 2,000 pounds (see Section 6) to full-scale (by multiplying by 2^3) yields a full-scale force of 16,000 pounds. To handle this load, 5/8-inch trade-size galvanized steel lift chain, with a maximum working strength of 22,000 pounds, was chosen. Any stronger chain would require a link size that was prohibitively large for a reasonably sized chain pocket.

The lift bridle is comprised of four lengths of wire rope each connected to a corner of the forward assembly (see Figure 49), and each meeting in a central junction above the center of the forward assembly. The geometry of the bridle is such that the central junction of all four wire rope cables is 48 inches above the top of the end longitudinals. The wire rope is ½ inch diameter galvanized steel wire rope with a minimum breaking strength of 21,400 lbs. With this geometry and the strength of the wire rope, the overhead factor of safety is for the entire device including Fast Sweep boom (4,000 pounds) was calculated to be 5.3 using a static analysis.

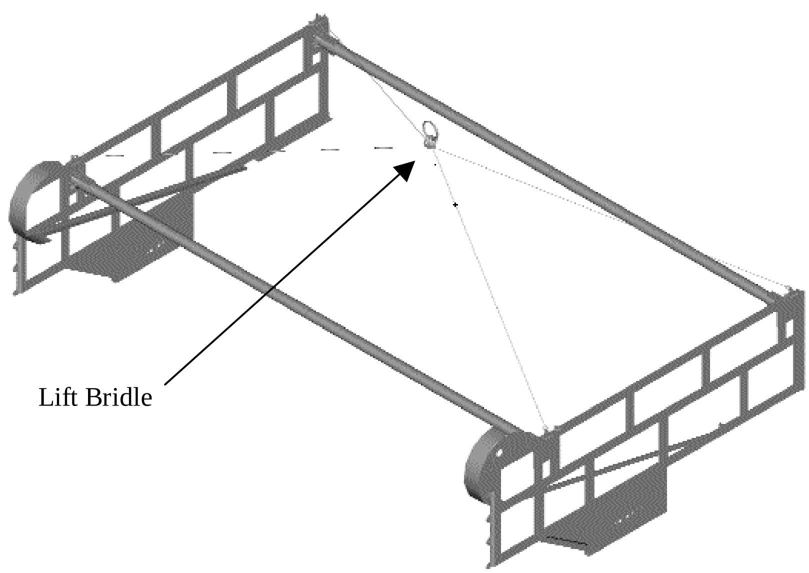

Figure 49. Full-scale lift bridle (basic forward assembly shown for clarity).

The full-scale racking cables are the same as for the half-scale model ; ¼-inch wire rope with a 7,000 pound working load capacity. Though not calculated explicitly, it is assumed that cable loading is an order of magnitude less than the maximum forward assembly forces, for example, the chain force at 16,000 pounds).

Tow bridles for the full-scale system were designed from the same wire rope used for the full-scale lifting bridle. This ½-inch diameter wire rope has a 21,400-pound working load capacity. Using the maximum tow force per end longitudinal of 10,400 pounds, (Froude-scaled from test measurements) and assuming the tow force is split evenly among the four bridle cables, the factor of safety is estimated to be about 8.

Flotation was scaled up from the half-scale to the full-scale system. The same fabric is proposed for full-scale as was used for half-scale. Side floats and forward float however, should contain at least two air bladders. Inclusion of bladders would reduce the risk of catastrophic flotation failure. Finally, the stainless steel fastener system worked well with the half-scale system. The fastener system would be the same as that used for the half-scale system but with higher-strength bolts.

Weight Scaling

The full-scale design is not an exact scaled replica of the as-built, half-scale system. Although the basic structural square tubing was geometrically scaled, some components, such as the hydrofoil mounting plate, forward float brackets, and compression member center assemblies were the same thickness or lengths at both scales. Sheet aluminum of ¼-inch thickness was intended for use at full-scale for the hydrofoil mounting plate. It was also used at half-scale because ⅛-inch thick sheet was not deemed structurally adequate with numerous drilled holes for hydrofoil mounting. The full-scale forward float brackets were intended to be made from ⅛ inch aluminum sheet and were made the same thickness at half-scale because 1/16 inch thick material was not deemed structurally adequate. The compression member center assemblies were designed to be 24 inches long on both scales because a length any longer was determined to be unnecessary.

Prior to the engineering analysis of the forward assembly and resulting addition of structural stiffeners, the complete full-scale design was 69.31 pounds lighter than the half-scale system as built, at full-scale size (see Table 5). As perfectly scaled weight would be difficult to achieve at the full-scale with incorporated modifications from the as-built, half-scale system, such a deficit was desired. When the engineering analysis determined that additional structural members were required at full-scale, the weight deficit would be closed up. The addition of the structural stiffeners was 321.44 pounds. This resulted in the full-scale system being 252.13 pounds heavier than the half-scale at full size. A complete listing of full-scale weights is given in Table 6.

Table 5. Weight comparison of half-scale model and full-scale design.

Components	No.	Half-scale wt. (lbs)	Half-scale wt. at full-scale (lbs)	Full-scale wt. (lbs)
End Longitudinals	2			
Base Frame	1	34.25	274	274.01
Forward Float Shield	1	3.714	29.712	17.78
Extension Plate less hyd. Mount plate	1	9.51	76.08	49.15
Hydrofoil Mount Plate	1	13.93	** 55.72	41.25
End Longitudinal Combined Stiffeners*				160.72
Total for one End Longitudinal		61.404	435.512	542.91
Total for both End Longitudinals		122.808	871.024	1085.82
Intermediate Longitudinals	11	68.63	549.04	367.86
Forward Compression Member	2			
Forward Compr.Tube	2	23.35	186.8	208.3
Forward Center Assembly	1	10.08	** 40.32	14.93
Forward Float Brackets	5	23.05	184.4	92.14
Compr. Locking Pins	4	1.6	12.8	12.79
Total		58.08	424.32	328.16
Aft Compression Member	2			
Aft Compression Tube	2	23.35	186.8	208.3
Aft Center Assembly	1	10.08	** 40.32	24.15
Compr. Locking Pins	4	1.6	12.8	12.79
Total		35.03	239.92	245.24
Hydrofoil	1	30.99	247.92	557.27
Total		**215.918**	**2332.224**	**2584.35**

NOTES:

1. * End longitudinal stiffener.
2. ** Hydrofoil mount plate, same thickness (¼") half-scale and full-scale. Half-scale to full-scale multiplier is 4.
3. Center assemblies are 24" in length at both half-scale and full-scale.
4. Full-scale weights taken from AutoCAD model.
5. Half-scale intermediate longitudinals 1 ¼" sch. 40 alum. pipe; full-scale intermediate longitudinals 2 ½" sch. 40 alum pipe.
6. Hydrofoil weight includes all flanges
7. Half-scale hydrofoil weight from Dugan (2000).

Table 6. Full-scale system weight.

Components	Weight In Air (lbs)
End Longitudinal (total 2)	1085.82
Forward Comp Member	328.16
Aft Comp Member	245.24
Intermediate Longitudinals (total 11)	367.86
Hydrofoil	557.27
Submersion Plane Fabric	92
Baffle Fabric	195
Fast Sweep Boom	1100
Total Weight	**3971.35**

IX. CONCLUSION

The goal of this investigation was to develop a high-speed oil skimming system for VOSS-type application alongside a buoy tender. The study brought established submergence plane technology to a specially designed device modifying an existing oil collection system. Basic performance, configuration and logistical features were established. An initial full-scale design was determined and a 1/5-scale model was built and tested successfully in a wave/tow tank and in the field. The full-scale design was revised, and a half-scale model was built and tested at Ohmsett for hydrofoil performance and oil retention capability. The hydrofoil performance was excellent – it solved the submergence plane rise problem at all speeds possible at Ohmsett. Oil containment was excellent at two knots, but dropped to about 60 percent at a convenient vessel operating speed of 3 knots. A full-scale design resulted from evaluation of the half-scale model. The full-scale design was then analyzed structurally.

Recommendations for Full-Scale System Prototype

Although half-scale testing of the system showed great potential at full-scale, the system would benefit from further research and development. The major items of concern are:

- Bow-form turbulence,
- Modifying the Fast Sweep Boom to better serve as the aft barrier,
- Strength of the end longitudinals and compression members under dynamical loading,
- Resistance of the hydrofoil to upwards bending in waves.

Bow turbulence could be studied using a laser-doppler-velocimeter in a recirculating flume or tow tank. Reduction in turbulence production may be possible by alteration of the bow profile. Modifications could then be tested using the existing half-scale model at Ohmsett.

One change to the Fast Sweep boom that is easy to make is to add a heavy ballast chain to hold down the horizontal baffle. Modifying the stern to eliminate the inside corners (where vortices form) is a more extensive change but is at least confined to a limited portion of the boom.

Cascade booming across the containment region of the half-scale model could be investigated at Ohmsett. The intent would be to impede surface flow-through allowing an increase in gap bite, thereby catching more oil that would otherwise go beneath the system.

Further strength analysis using finite-element modeling would yield a more comprehensive understanding of stress distribution and would result in a more robust final design. The loads on the hard-structure being analyzed, however, are difficult to specify. The stress prediction would only be as good as the assumptions made regarding fabric tensions, wave velocity distribution, wave-system interaction, drag coefficients and critical masses.

Alternatively, it may be better to build a prototype based on the full-scale design presented here which has been optimized based on what is presently known. Strength considerations could then be addressed by fully instrumenting critical locations by means of strain gauges and load cells. The prediction of better oil retention at full-scale could be verified quantitatively in oil retention tests. Trials would take place at Ohmsett first, then in the field. Should system performance be good, consideration should be made to integrate the system into the USCG's spill response inventory.

Recommendations for Half-Scale System

The half-scale model tested unexpectedly well in oil collection and sea-keeping. Its performance merits consideration as a full-scale system in itself. As a full-scale system, it has great advantages over the full-scale system as designed herein. The half-scale system is several times lighter, and much more manageable on a buoy tender deck. The system is also is also easily transported. The entire device including boom can be loaded into a large cargo van. These qualities also lend it possible usefulness on smaller USCG vessels. Determination of its use as a full-scale system should begin with sea-keeping tests along an intended platform. An ideal platform may be the USCG's 55-foot aids to navigation boats.

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APPENDIX A
FROUDE SCALING PROCEDURES AND MODEL SPEEDS

Froude Scaling Laws

When physical modeling of systems where gravity and inertial forces dominate over other forces, such as viscous, or elastic, Froude scaling is used. This particular project was similar to naval architecture modeling; the system was subject to drag and inertial forces from fluid flow and gravity-restoring waves. Thus, Froude scaling was employed. The Froude number is defined as the ratio of inertia force to gravity force, or mathematically as

$$Fr = \frac{v}{\sqrt{(gL)}}$$

Where

v = the fluid velocity

g = the acceleration due to gravity

L = the comparative length parameter.

In order for one model to be a Froude-scaled model, the model and system must be geometrically similar and have identical Froude numbers.

To predict the full-scale behavior and define model-testing parameters, λ is defined as the ratio of the full-scale characteristic dimension to the model's characteristic dimension. Table A1 lists some useful Froude scaling relationships and their corresponding scale factors:

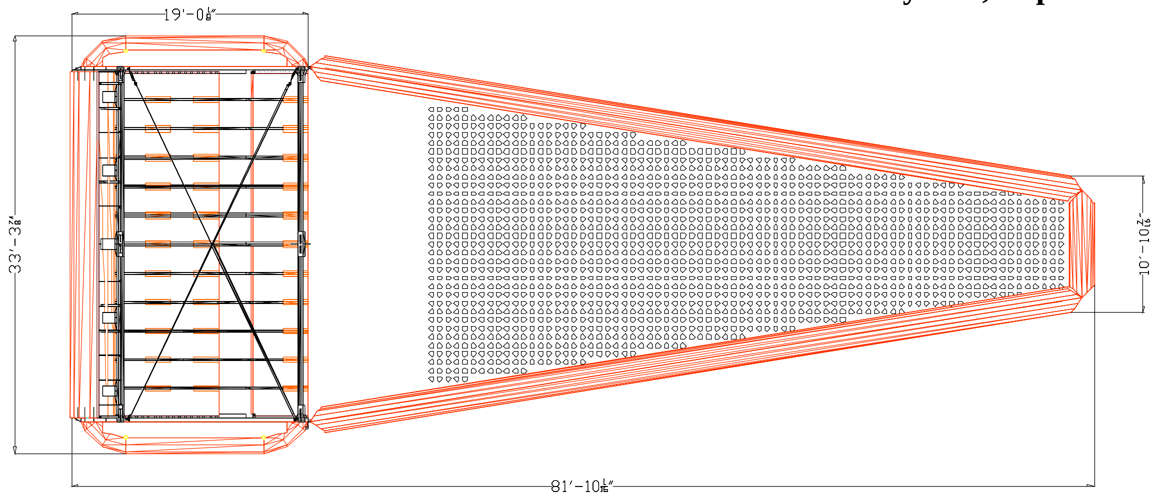
Model parameters and their associated froude scale factors.

Parameter:	Length	Area	Velocity	Force	Moment	Stress	Wave Height	Wave Period	Wave Length	Density
Scale Factor:	λ	λ^2	$\lambda^{1/2}$	λ^3	λ^4	λ	λ	$\lambda^{1/2}$	λ	1

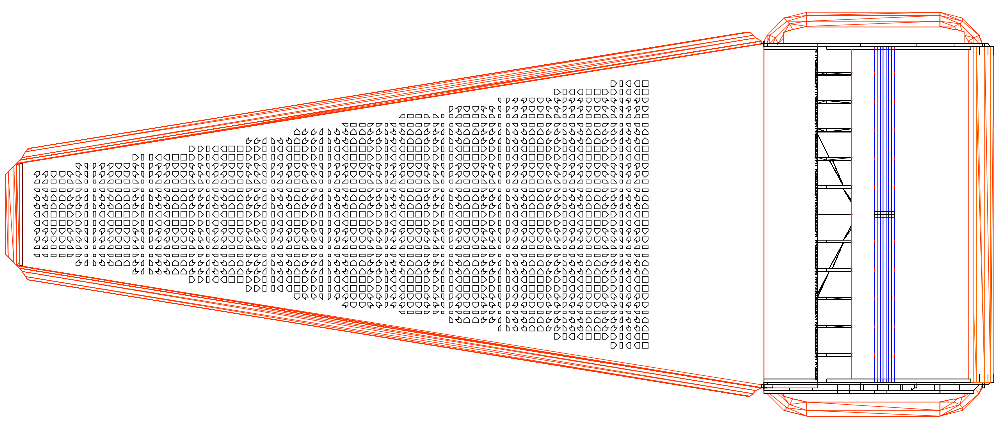
APPENDIX B

DRAWINGS OF FULL-SCALE SYSTEM

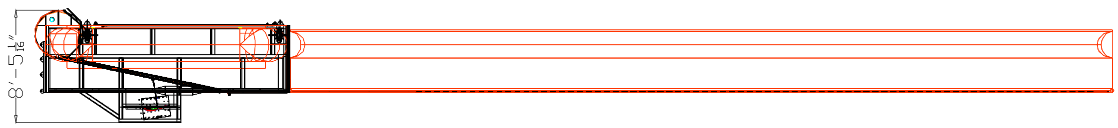
Full-Scale System, Top View



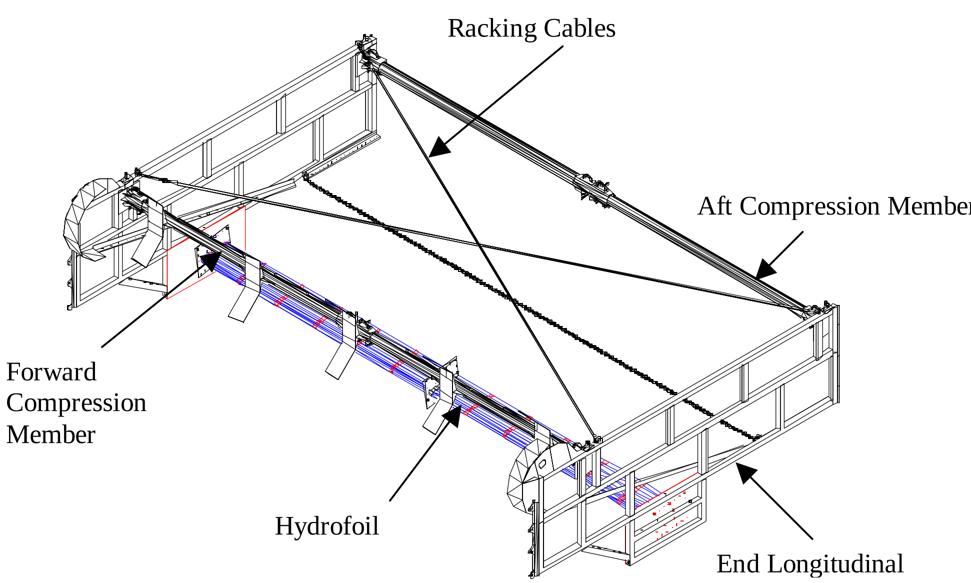
Full-Scale System, Bottom View

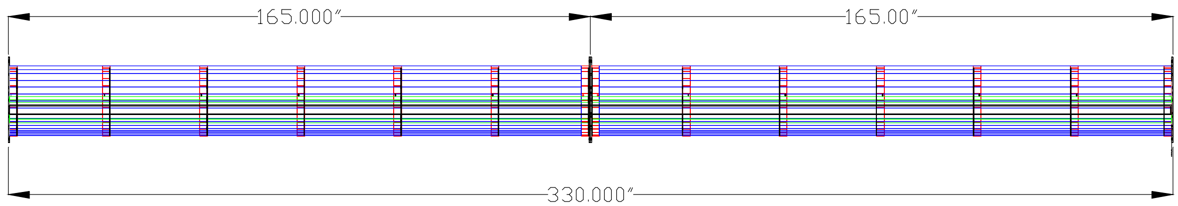


Full-Scale System, Side View

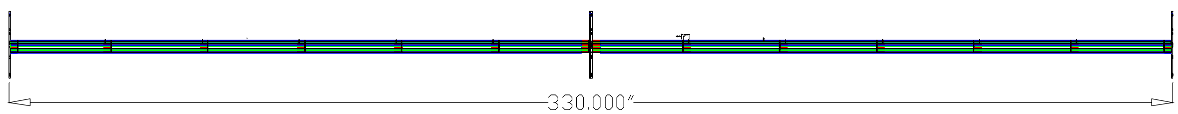


Rigid Assembly Major Components

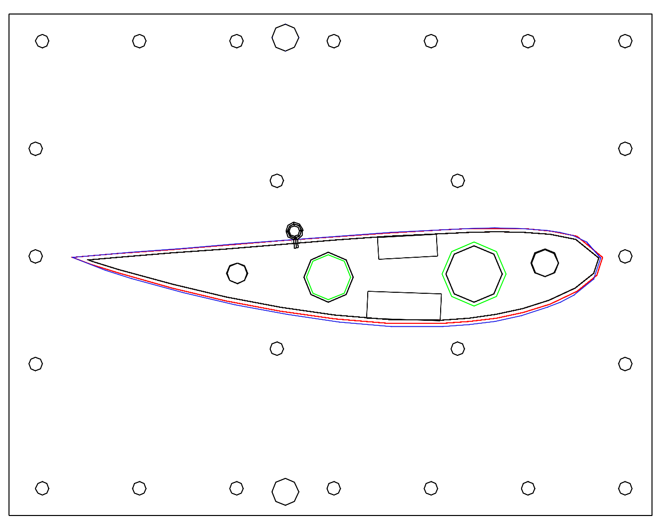




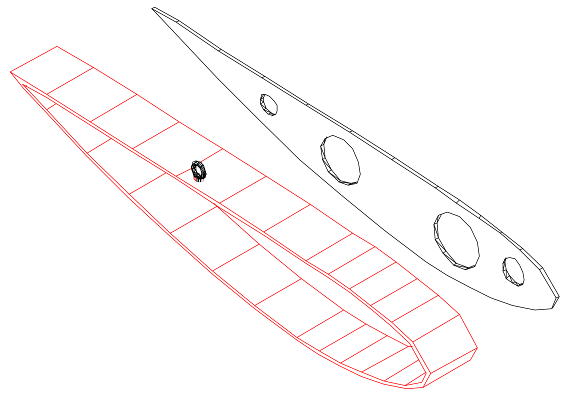
Hydrofoil, Top View



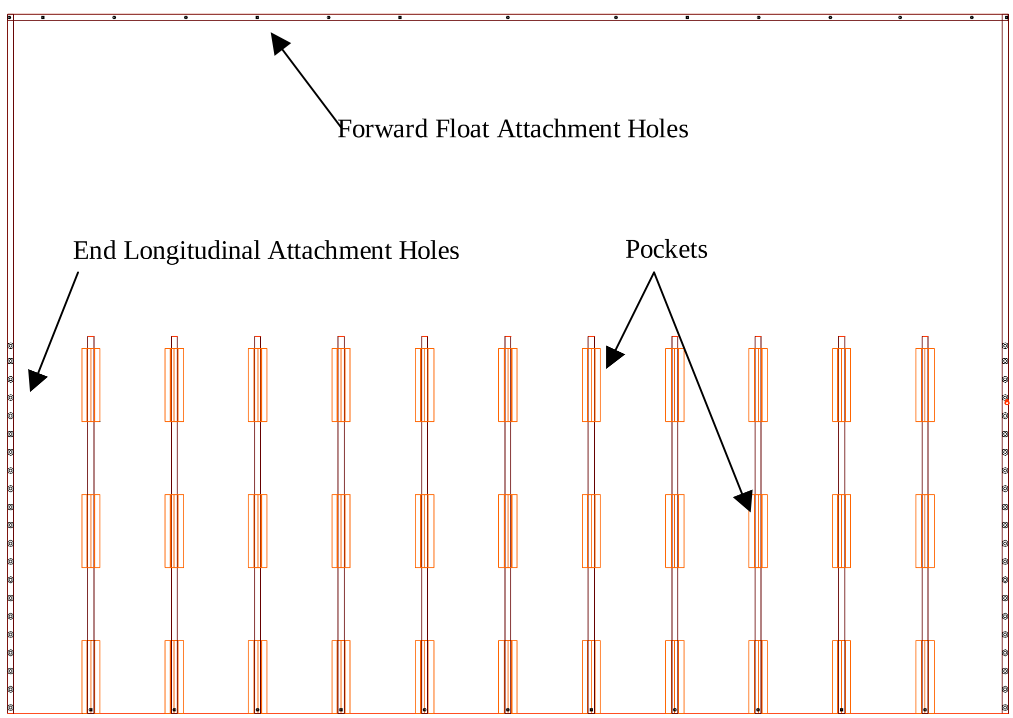
Hydrofoil, Front View



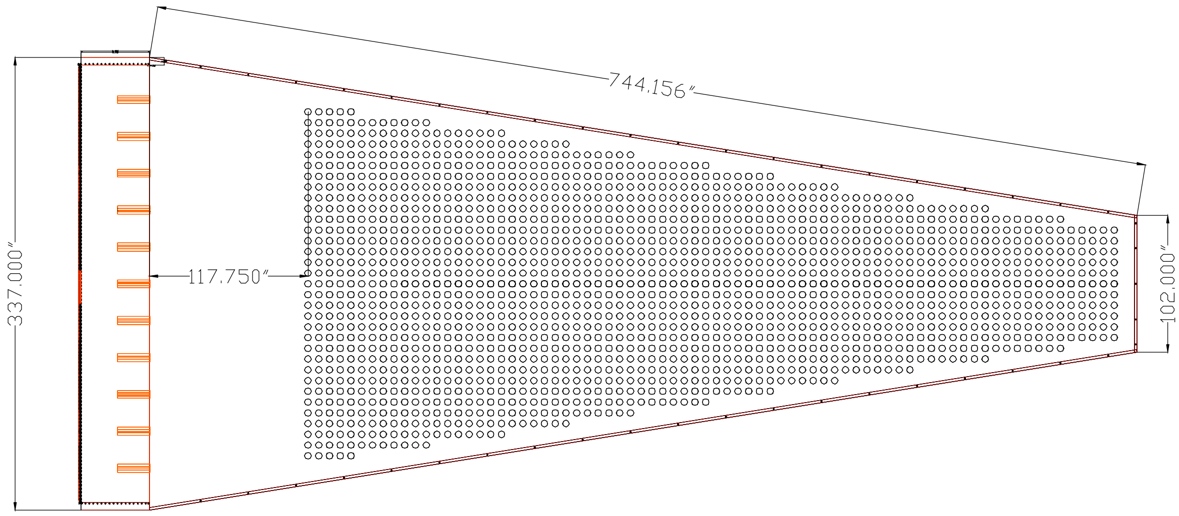
Hydrofoil, Base Flange



Rib Construction Detail



Submergence Plane, Plan View



Horizontal Baffle, Plan View

APPENDIX C
ENGINEERING ANALYSIS OF THE FULL-SCALE PROTOTYPE

Purpose

Finite element and strength of materials analyses were applied to the full-scale design to ensure adequate strength. Though the half-scale system had performed well structurally during the Ohmsett testing and served as the basis for the full-scale design, stresses double in going to full-scale. An iterative cycle of analyses and redesign took place at critical locations until sufficient strength was achieved. In the following discussion, the design obtained by doubling half-scale dimensions is analyzed showing overstressing; the components have the necessary strength. Forces used in the analysis were derived by two means:

- Froude scaling to full-scale tow forces recorded on the half-scale model

- Froude scaling to full-scale analytically-derived baffle drag forces calculated for the half-scale model.

Base Structure and Hydrodynamic Forces

Maintaining the submergence plane and gap geometry is critical to system performance. The forward assembly of the system is, therefore, rigidly framed by means of end longitudinals, compression members and a hydrofoil, which also acts as a compression-resisting member. Three possible major failure modes of the forward assembly are considered:

- Transverse failure of Forward Assembly
    i. Failure of Baffle Chain
    ii. Failure of Compression Members in buckling or shear
    iii. Failure of End Longitudinal due to out-of-plane bending
- Longitudinal failure of Forward Assembly in tension

The first possible mode of failure is a result of compression member or hydrofoil failure under buckling or shear or chain failure in tension due to drag forces on the baffle. The baffle chain forms a catenary and the resulting tension acts to pull the two end longitudinals together. A combination of two compression members, both forward and aft, and the hydrofoil resist the chain tension. Another possible mode of failure is the end longitudinals failing under the tension of the tow forces. (see Figure C-1)

The cables to the submergence plane counteract the hydrofoil negative lift distributed across its length, which is subject to upward lift. Thus hydrofoil and submergence plane act as a balanced unit applying very little net force to the forward assembly structure. Net end forces are a magnitude smaller than those of the two possible major failure modes, and as such, were not investigated. Figures C1 and C2 show the major loading upon the structure.

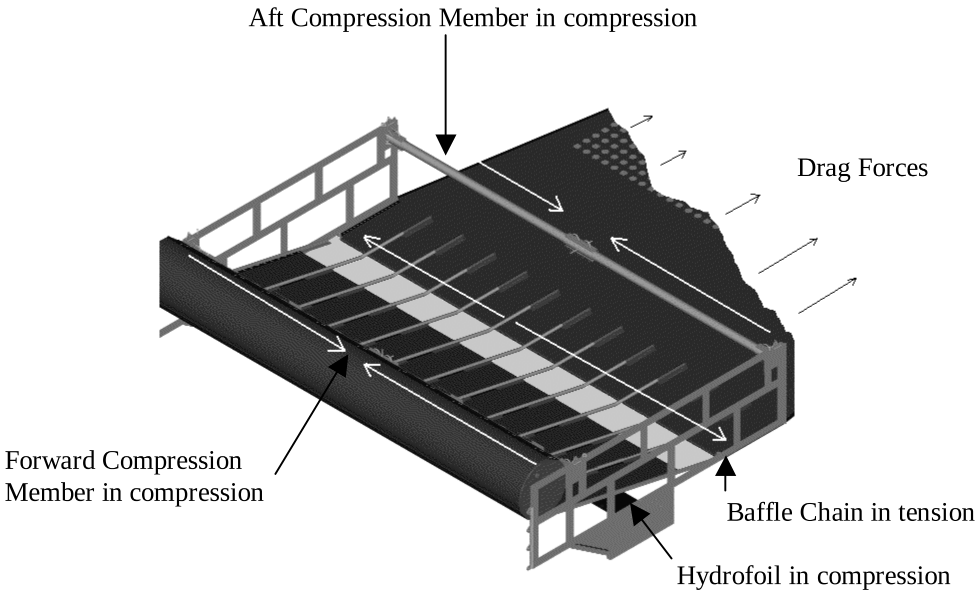


Figure C1. Transverse loading of forward assembly.

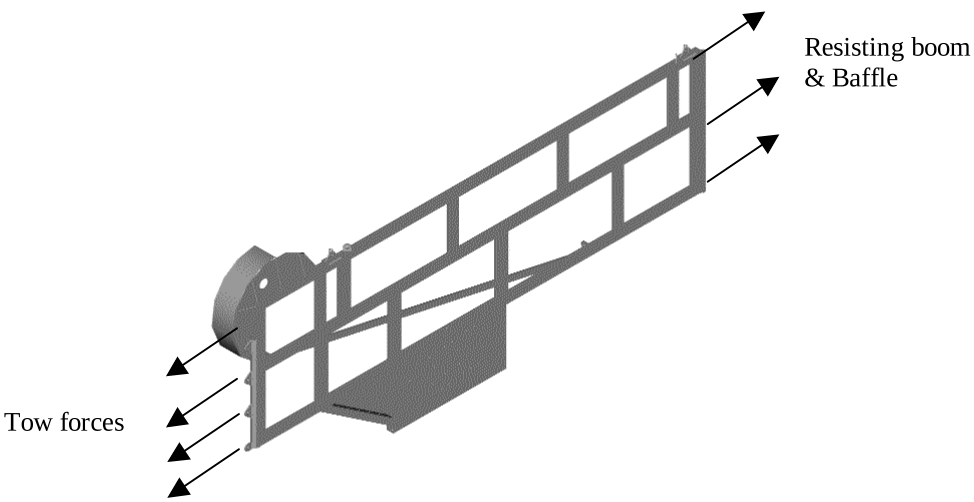


Figure C2. Longitudinal loading of forward assembly.

Approach

The analysis of both failure modes was conducted using a finite element approach. The computer program used was the *Designer* linear finite element analysis package by the Marc Research Corporation. This program makes it easy to change the mechanical and geometric properties of structural elements and provides the latitude to vary structure element complexity. It was desired to make the model three-dimensional, but simple. The model used contained only major horizontal and vertical structural items of the end longitudinals and compression members. Boundary conditions were selected and placed in combination and in locations on the model, which would represent a 10-knot full-speed survival operation.

Modeling

The computer model was a simplified three-dimensional version of the main structural components of the full-scale forward assembly and included end longitudinals, baffle chain, compression members and hydrofoil (see Figure C3). Each major structural component was divided into several elements. Mechanical values of the full-scale structural components, including area moment of inertia, were derived through the area property function of AutoCAD. These properties were assigned to their corresponding elements. The model contained 151 nodes and 179 elements. All elements, including compression members were modeled as thin beams, having major and minor axis moments of inertia and cross sectional areas. Modeling of the compression members and hydrofoil as thin beams instead of truss members was necessary because of the inherent bending loads that they would realistically be under, which would significantly contribute to their total stress. Also, they were required by the program to bend under stress to allow movement of the end longitudinals. The presence of the extension plate aluminum plating was considered in the cross-sectional properties of the extension plate-framing members.

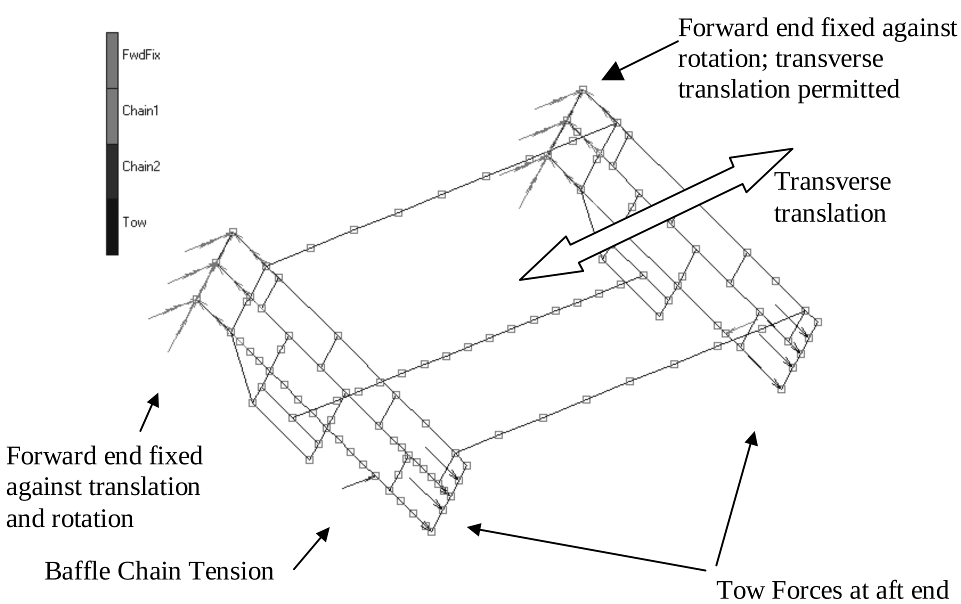


Figure C3. Computer finite element model of forward assembly.

The program required the input of boundary conditions whereby certain points of the structure were fixed to prevent rigid body motion. The forward end of the structure was fixed to prevent rigid body movement in the following manner: the forward-most member of the port end was fixed against rotation and translation and the forward-most member of the starboard end longitudinal was fixed against rotation, and all translation except in the transverse direction. As indicated in Figure C3, these conditions (fixities) were applied to the forward end instead of the aft end. Since the forward end was connected directly to the tow cables, its tendency to translate or rotate would be far less than at the aft end of the structure. Translation in the structure's transverse direction at the starboard end longitudinal forward connection would allow bending of the forward compression member. The aft end of the full-scale forward assembly would be connected to the Fast Sweep boom. Although the boom's connection at high speed would undoubtedly offer some additional rigidity to the structure, this was considered negligible. Also, the transverse tension in the submergence plane fabric, which naturally tends to pull the two end longitudinals together thus placing compression forces in the compression members, was deemed much less than the axial forces in the chain and compression members.

Since the forward end of the structure was largely fixed, the tow forces were applied to the aft end at locations corresponding vertically to the actual locations on the forward end. Tow force loading consisted of 10,400 lbs. per end longitudinal. This force was obtained from a trend line developed from half-scale model recorded load cell data, Froude scaled to full size (see section 6).

The baffle chain tension was applied to the model as point loads with locations corresponding to the chain connection to the end longitudinals. The tension force of 16,500 pounds was from the analytical approach and equation for baffle drag (described in the HALF-SCALE MODEL section) using full-size dimensions.

Since bending and tension stresses predominated, a criterion was established to determine if computer-generated results were acceptable. The allowable tensile and bending strength of the aluminum was taken to be 35,000 pounds per square inch (Lindeburg, 1999). A minimum factor of safety of 2.0 was also established for all structural members including hydrofoil and compression members. This meant that working stresses within the loaded system could not reach more than 17,500 pounds per square inch.

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Results

The *Designer* output file produced values for moments about the major and minor axis, and the axial tensile or compressive force. Two values of these three entities were shown at two separate points along the length of each element. The values were interpreted into an equivalent stress using the formula:

$$\sigma = \frac{M * c}{I} + \frac{P}{A} \qquad \text{(C1)}$$

where

σ = Maximum stress of the material (lb_f / in^2)
M = Moment about each axis (lb_f*in)
c = Maximum distance from neutral axis of member cross-section to extreme bending fiber (in)
P = Maximum axial force, compression or tension (lb)
A = Cross-sectional area of member (in^2).

A calculation was conducted for each element to check to see that the maximum allowable stress was not exceeded. The calculation consisted of taking the absolute values of maximum moment, *M*, and the axial force, *P*, and placing them into the equation.

As expected, transverse failure of the forward assembly occurred due to out-of-plane bending of the end longitudinals as can be seen in Figure C4. Localized bending in the vicinity of the chain connection caused stresses far beyond the yield strength. The elements situated at and adjacent to each of the two chain connections experienced extreme bending stress on the order of 125,000 pounds per square inch, nearly 4 times the allowable bending stress. Transverse displacement at the aft end of the end longitudinals reached 11.5 inches. The mid-strength member of the extension plates also experienced stress exceeding the allowable stress.

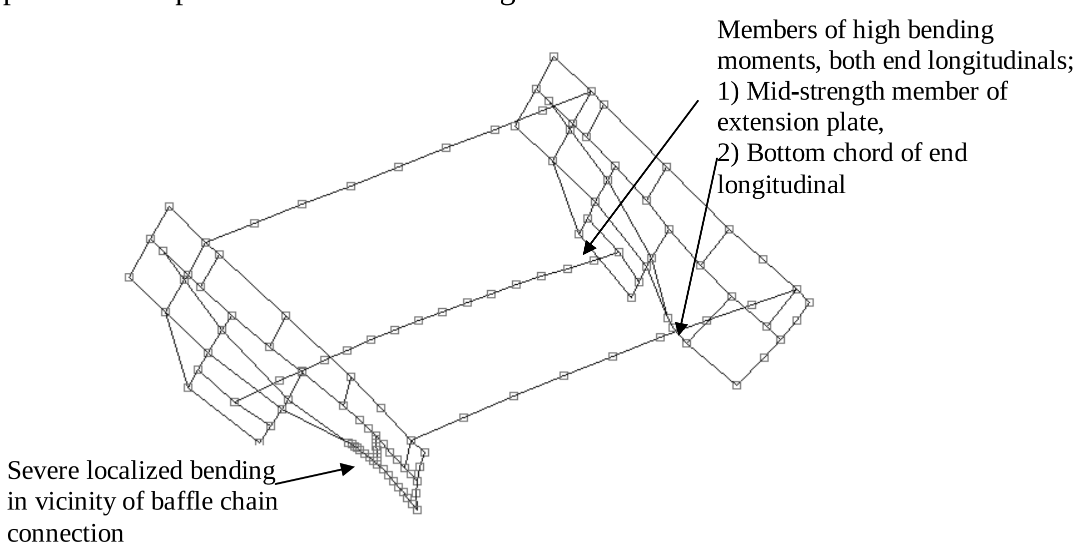

Figure C4. Finite element model transverse failure deflection results (results magnified 5 times).

The calculated stresses in the remainder of the end longitudinals, the hydrofoil and compression members were below the allowable stress. It was apparent that additional rigidity would be necessary along the aft end of the end longitudinals' bottom chord. The area would have to be stiffened; yet only minimal additional weight could be added to the end longitudinals due to weight issues. Various shapes with area moments of inertial much larger than that offered by the 3-inch by 3 inch by ½ inch thick aluminum square tubing were considered for augmenting the bottom chord.

Design Strengthening and Analysis

Several trial Tee-section structural members were analyzed as an augmentation to the bottom 3 inch by 3 inch by ½ inch square tubing. The mechanical properties of the various Tee sections were assigned to the bottom square tubing members of the end longitudinal and the framing members, particularly the mid-strength member of the extension plates. The Tees were chosen due to their high-yield of area moment of inertia per cross-sectional area. Connected to the square tubing by continuous welds as shown in Figure C5, the section would augment the portion of square tubing into which the baffle chain was connected.

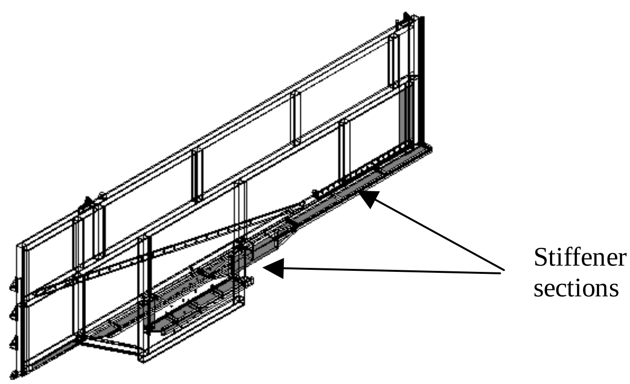

Figure C5. Stiffener sections added to end longitudinals as a result of the engineering analysis.

The process of proposing various Tee sections in both high-stress locations was iterative. The larger the Tee section used, the greater the concentration of stress in that particular location. The section modulus of the given section would have to be sufficiently high enough to achieve a stress factor of safety of over 2.0. The aim was to find the smallest cross-sectional Tee that could be found since weight considerations were paramount (next to strength). A small cross-sectional member that satisfied the required moment of inertia could not be found from stock shapes. Instead, several custom-sized sections were used which minimized cross-sectional area and thus weight, while providing maximum area moment of inertia. These sections were used in various combinations of lengths and locations in the high-load regions. The optimal solution was found to be a combination of T section stiffeners and gusseted corners. These stiffeners when combined with the attached square tubing members of the end longitudinals, provided the required rigidity against bending.

When the model was run with these new members, the highest stresses of the structure, those adjacent to the chain connections were well below the allowable stress. The total stress immediately adjacent to the chain connection was 14,430 pounds per square inch. The factor of safety against allowable stress for the member with the highest stress was 2.171. Displacement of the end longitudinals had also been greatly decreased. Instead of the previous 11.5 inches of transverse displacement at their aft ends, the end longitudinals displaced only 3.14 inches.

Compression Members and Hydrofoil

As part of the model, results for the compression members were also computed. Results showed that while the aft compression member was in compression, the forward compression member was actually in tension. Total stress for the aft compression member was 2335 pounds per square inch, well below the allowable stress of 35,000 pounds per square inch. Total stress for the forward compression member was 3952 pounds per square inch. Although further away from the applied chain load, the forward compression member’s stress was larger than the aft compression member. This was due to the forward compression member having a bending moment roughly twice that of the aft compression member. As expected, the output axial force component for each compression member did not vary along its length as the bending moment output did.

The hydrofoil component of the model was found to take the largest compressive load at 16,441 pounds. This was expected, since the hydrofoil was the closest transverse component to the baffle chain. Total stress including bending and axial was calculated for each axis of the hydrofoil. Bending in the vertical plane was found to govern, as expected. Total stress was found to be 13,300 pounds per square inch, well below the maximum allowable stress of 35,000 pounds per square inch.

Supplemental Calculations

Due to their slenderness, the compression members and hydrofoil were also analyzed for buckling using the same Euler approach as was used on the one-half-scale model. Although the computer model had accurately determined structure loading and deformations for each component as determined by component interaction in a large structural system, calculations to determine catastrophic failure of the slender members was deemed appropriate as a check. The critical Euler buckling force for each compression member was 18,620 pounds; for the hydrofoil, 54,880 pounds. Thus, each compression member was found to be capable of resisting the entire baffle chain load of 16,500 lb with a factor of safety of 1.13. The factor of safety against hydrofoil buckling using the top speed compressive force was found to be 3.34. In actuality (from the finite element analysis) the forward “compression member” is in tension. Using the finite element compressive force result of 4,726 pounds for the aft compression member, its factor of safety was 3.94.

Full-Scale Design Modifications

All required structural modifications were made to each end longitudinal. The compression members and hydrofoil were found to be more than adequate structurally. The composite stiffener sections found to offer sufficient rigidity to the forward assembly structure were incorporated into the bottom member of each end longitudinal. Additional gusset plates were added at other critical locations to increase area moment of inertia. The total weight of all stiffeners added to each end longitudinal was 160.72 lbs. This increased the weight of the full-scale forward assembly by 14.2 percent and increased the weight of the full-scale system 10.8 percent above the weight of the half-scale system scaled to full-scale. Thus the system could be well assured of surviving 10 knots and would weigh just over 10 percent of its intended design weight.